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JUN 2 4 1980

NAS 1.55:2106



NASA Conference Publication 2106  
DOE Publication CONF-7904111

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# Large Wind Turbine Design Characteristics and R&D Requirements

A workshop held at  
Lewis Research Center  
Cleveland, Ohio  
April 24-26, 1979



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NASA Conference Publication 2106  
DOE Publication CONF-7904111

# Large Wind Turbine Design Characteristics and R&D Requirements

Seymour Lieblein, Editor  
Technical Report Services  
Rocky River, Ohio

A workshop held at  
Lewis Research Center  
Cleveland, Ohio  
April 24-26, 1979

## FOREWORD

The Department of Energy (DOE) is committed to a vigorous research and development program to develop useful electric power from the wind. The DOE Wind Energy Program\* contains the major elements of program development and technology, farm and rural (small) systems, 100 kW-scale systems, megawatt-scale systems, and large scale multi-unit systems. Considerable effort has been exerted on these programs which are now yielding significant results and information.

In order to enhance the communication and discussion of these results, the DOE has sponsored a series of six workshops on specialized wind energy topics during 1979. The specific objectives for the series of workshops were to: (1) present the progress and significant results of ongoing projects sponsored by DOE, (2) provide a forum to facilitate the exchange of new ideas and information, and (3) provide an opportunity for in-depth discussion of specific issues confronting wind turbine developers as the technology moves closer to the goal of commercialization.

The 1979 workshops were organized under the auspices and with financial support of the Wind Systems Branch of DOE. Each workshop was coordinated by people intimately concerned with the major topics involved. Details of the six 1979 workshops are as follows:

1. "SWECS 1979: A Workshop on R&D Requirements and Utility Interface/Institutional Issues," February 20-22, 1979, Boulder, Colorado. Coordinator: Darrell Dodge, DOE Rocky Flats Plant, P.O. Box 464, Golden, Colorado 80301. Proceedings: DOE Conf-790243.
2. "Economic and Operational Requirements and Status of Large Scale Wind Systems," March 28-30, 1979, Monterey, California. Coordinator: Edgar A. DeMeo, Electric Power Research Institute, Box 10412, Palo Alto, California 94303. Proceedings: DOE Conf-790352.
3. "Design Characteristics and R&D Requirements for Large Wind Turbine Systems," April 24-26, 1979, Cleveland, Ohio. Coordinator: Patrick Finnegan, NASA-Lewis Research Center, 21000 Brookpark Rd., Cleveland, Ohio 44135. Proceedings: DOE Conf-7904111. (NASA CP-2106)
4. "Wind Power for Agriculture," May 15-16, 1979, Ames, Iowa. Coordinator: Thomas McCormick, Iowa State University, Ames, Iowa 50011. Proceedings: DOE Conf-7905109.
5. "Wind Energy Innovative Systems Conference," May 23-25, 1979, Colorado Springs, Colorado. Coordinator: Irwin Vas, Solar Energy Research Institute, 1536 Cole Boulevard, Golden, Colorado 80401. Proceedings: DOE Conf-790501.

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\*Wind Energy Systems Program Summary. U.S. Department of Energy, Office of Solar, Geothermal, Electric And Storage Systems. DOE/ET-0093, Dec. 1978.

6. "Conference and Workshop on Wind Characteristics and Wind Energy Siting,"  
June 19-21, 1979, Portland, Oregon. Coordinator: Larry Wendell, Battelle  
Pacific Northwest Laboratory, Battelle Boulevard, Richland, Washington  
99352. Proceedings: DOE Conf-790665.

This series of workshops will lead to a final meeting, the Fourth Biennial Conference and Workshop on Wind Energy Conversion Systems, scheduled for late October, 1979. The program for this conference will include R & D activities being sponsored by DOE, discussions of the results of the workshops held earlier in the year, and descriptions of ongoing international activities. Issues and problem areas related to wind energy will be discussed.



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## INTRODUCTION

A major phase of the DOE Wind Energy Program involves research and development on large size wind turbine systems that can eventually be integrated into utility networks. Both horizontal and vertical axis machines are considered in this activity. Elements and results involved in this program served as the major topic of the third workshop in the 1979 series.

The workshop on Large Wind Turbine Design Characteristics and R & D Requirements was held in Cleveland, Ohio, April 24-26, 1979. The workshop was conducted by the NASA Lewis Research Center, which is responsible to DOE for the development of large horizontal axis wind turbines. Sandia Laboratories was responsible for the vertical axis portion of the workshop.

The specific objectives of the Cleveland workshop were to: (1) describe the characteristics and development status of current large wind turbines, (2) identify the technical problems that must be solved to achieve the cost goals; (3) identify and discuss promising solutions; and (4) describe the R & D effort required to demonstrate the feasibility of the proposed solutions.

The workshop consisted primarily of detailed technical presentations on large wind turbine R & D activities sponsored by DOE. Information on large wind turbines being developed by several private organizations was also presented, and large wind turbine projects in Denmark, Sweden, and West Germany were reviewed. Panel discussions after each major session provided an opportunity to discuss issues and problems. Workshop outline and specific topics can be ascertained from the CONTENTS.

The workshop was sponsored by the DOE Wind Systems Branch. Workshop coordinator for Lewis was Patrick M. Finnegan; Joseph M. Savino was sessions chairman. Attendees numbered 168, with 115 from industry, 31 from government laboratories and DOE, 21 from foreign countries, 18 from universities and 4 from utilities.

This document presents the proceedings of the workshop. It contains both prepared formal papers and edited transcriptions of panel discussions and questions and answers for the individual papers.

OVERVIEW OF  
FEDERAL WIND ENERGY PROGRAM

Daniel F. Ancona  
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Washington, D.C.

This discussion will provide a brief overview of what the Federal Wind Program is today, what the objectives are, and what strategies are being followed. Some of the changes in the program structure and some of the additions to the program will also be included. There will be mention of upcoming organizational changes, and some budget items will be covered, with particular mention of some recent significant events regarding new approvals. Lastly, there will be a question and answer session after the formal remarks.

First of all, the overall objective of the Federal Wind Program is to get the Government out of the wind energy business. It is our desire to develop machines that are reliable and truly cost competitive. In this way, industry will be moved to the point where it can produce wind machines and sell them to utilities and to private individuals so that significant quantities of electrical energy can be captured from the wind.

The general thrust of the program is to start with mission studies and application studies. As you know, we have come a long way from some of the early machines to the machines that are now in use. It is hoped, through the Wind Energy Program, to go a long way beyond that.

The strategy used to achieve the objectives is based on the elements shown in figure 1. The first activity, Application Studies, will be combined with Legal, Social and Environmental Studies and some long range planning activities to comprise a new program element that is called Research and Analysis. This new program structure will be seen more frequently in the publications such as program summaries that emanate from Washington.

Wind Characteristics will continue to stand alone as a very important and significant part of wind energy research. Technology Development also will continue as a separate element, since it is an important area that feeds into all of the Experimental Systems Development programs, which is the next item. This category includes the small machines, the intermediate sizes, and the large machines for the various applications.

There will eventually be another element added to the bottom line to show Applications and Demonstrations. This will be a new line item referring to future marketing activities that we feel will have to be done. In fact, budgets are being planned for this new element which will be added to the program. The first budget reflecting this activity will be in fiscal year 1981.

The principal items covered in each of the major program elements are listed in figure 2. Some of the activities involved in these items will now be discussed.



## MISSION AND APPLICATION STUDIES

There have been a number of studies that look specifically at applications for wind turbines (fig. 2(a)). For example, the New England Gas and Electric System was chosen as the subject for a study of the economics and some technical issues of implementing wind turbines in an existing utility grid. The NEGEA service region is shown on the map of figure 3 with some of the wind characteristics for that area. Other studies have been done for such areas as Hawaii and Michigan. In fact, we also looked at more specific applications. For example, a study was done by Aerospace Corporation on the application of wind power to the California aqueduct system. There are studies under management of the U.S. Department of Agriculture for applying wind power in farm applications. Figure 4 shows one of the machines that is running today on a farm.

There are several studies, started recently, that fall into this general area of missions and applications. Two of them, in particular, involve working with the Tennessee Valley Authority to examine the implementation of a significant penetration of wind turbines into that general area. Specifically, the study is looked at the operational problems associated with interconnecting large wind turbines on the TVA grid with their generation mix.

Another study just recently started is examining the issue (economic factors in particular) of applying wind energy conversion systems in rural electric co-op activities. More of such studies will be initiated as the program develops.

Testing of experimental wind turbines remains a major part of the program. You may recall the 17 candidate sites that were picked from a group of about 55 proposals several years ago. A number of these sites now actually have machines operating on them. The map of figure 5 shows the current 17 sites. According to our current planning, there may be another round of site selections this summer for future machines.

## LEGAL, SOCIAL AND ENVIRONMENTAL ISSUES

Legal, social and environmental issues (fig. 2(b)) are a very significant part of the Wind Program. Legal concerns in a number of areas have been encountered; land use is one. There has been concern about where wind turbines might be sited, e.g., the aesthetics of wind turbines and the public reaction to their visual impact. For example, figure 6 shows a picture of a MOD-2 mounted on top of a hill. Actually, there are three machines along a ridge spaced typically about a quarter of a mile apart. The initial reaction is that there probably isn't any serious visual impact involved, but it is an issue that must be considered in siting machines.

Public safety is a social question involving the zoning of the area around a wind turbine. When a wind turbine is installed on a site, there is an area underneath the machine called a footprint, which must be owned outright. Such a piece of land might be of the order of an acre in size. However, there is an additional area around the base of the machine that may

be used for agricultural or other purposes, but not for dwellings. This area might be called a safety zone. It is probably analogous to the area underneath or in the vicinity of power lines. It is an item of concern at this time because land use problems are a part of life today, especially the approval of environmental impact statements.

Machines are often located in isolated areas. Figure 7 shows the MOD-OA at Culebra, Puerto Rico, which is a pretty isolated area. In such cases, there isn't much of a land use problem.

Another potential environmental impact from wind turbine is electromagnetic signal interference. An example is the TV interference problem. This is actually a reflection situation where the video portion of television signals may be reflected by rotating blades (fig. 8). This can cause an interference for people living in the immediate proximity of the wind turbine. We think we have solutions to this kind of problem.

#### WIND CHARACTERISTICS

One of the major aspects of the Wind Characteristics program (fig. 2(c)) is the determination of the wind resources in the United States. In the map of figure 9, the areas that have the highest wind are shaded, followed by lighter shading, and the more benign areas are unshaded. This is a very broad estimate of the resources that are available. Consequently, it is necessary to establish in more detail what resources are available in specific areas. As a prototype for that kind of a study, we examined the Northwestern region of the United States and acquired as much wind data as possible. The pins in the map of figure 10 represent sites where some kind of wind data was acquired. Sources are the traditional sites such as airports, NOAA stations (where very good detailed data are obtained), and other less obvious sites such as forest service fire towers. Many different kinds of information were compiled in an effort to determine the wind resources in more detail in that region. The study of the Northwestern region is about to be published. This work will serve as the basis for additional studies in the rest of the United States. In fact, requests for proposals to do other regions of the U.S. were issued, and the replies are in.

The wind resources studies and their attendant mathematical modelling are a very important step leading to the identification of a potential site for a wind turbine. An anemometer tower such as the one shown in figure 11 can then be placed at the potential site to evaluate in detail what the resources are. It is important to recognize that anemometer towers are very expensive, and locations must be carefully selected. The wind area studies provide a necessary tool to predict where the best resources are.

In regard to wind resources, it should be noted that siting handbooks will be made available. One for small wind machines has already been published. Another one is in the mill for large machines. Hopefully, these will help utilities and private individuals to properly site their machines.

Wind forecasting is the last area that should be mentioned. There is a lot of information available. It is basically a matter of organizing the information and translating it into a language that wind system users can understand.

#### TECHNOLOGY DEVELOPMENT

Activities in the Technology Development area (fig. 2(d)) can be illustrated by the familiar MOD-0 machine at the Plum Brook station. Figure 12 shows a photograph of the present installation. A lot of things have changed on that machine. As part of the technology development program, NASA has been doing extensive modifications of the MOD-0 machine. The blades in the figure are not the original blades. The one shown is a forerunner of the tip control teeter arrangement that is being tested on the machine now. Hopefully, it will be seen during the inspection tour.

The whole thrust of this program is, through component development and analytical studies, to learn the basic phenomena involved, so that the wind can be harnessed in a cost-effective way.

It should be mentioned, that since this is a workshop, one of the things to be determined is whether there are some things that we are not doing. In recalling the Dynamics Workshop that was held here about a year or so ago, there was considerable discussion at that meeting as to whether the MOD-0 machine should be run in a free yaw mode; that is, instead of using the motors to keep the machine pointed into the wind, to let it run free. NASA was almost challenged to try that mode of operation. As a result of the workshop, it was tried, and after a lot of analytical study, it was determined and verified on the MOD-0 that free yaw worked and that the blade wouldn't wrap around the tower.

A lot of other things were also tried, such as an upwind rotor and a downwind rotor and much significant data will be reported today. There are many things that are being done in technology development, and it is hoped that this meeting will reveal other things that should be done.

Components are another major element in technology development. Figure 13 shows a 150 foot blade that was built to evaluate new transverse filament fiberglass-reinforced plastic materials and manufacturing techniques for large blades. This type of blade, which had never been built before and which some people called the world's largest fiberglass fishing pole, was successfully fabricated and tested.

Figure 14 shows another blade that is currently flying on the MOD-0 machine. This blade, which was built here at NASA, uses a potentially very inexpensive construction technique. The spar of this blade is built from what might be called a telephone pole or utility pole. The airfoil shape is built up from wood ribs covered with razerback cloth. The idea here was to not only obtain a low-cost construction technique, but also to construct a blade that could easily be changed to a tip control configuration. More will be heard about this today.

The technology program is not limited to horizontal axis machines. Much will be presented in this conference about vertical axis machines. Figure 15 shows the Darrieus Machine at Sandia Labs, which is definitely one of the promising areas. There are indications now that the Darrieus vertical axis design, in general, may be able to compete directly with the horizontal axis machine. Thus, this area of the program is currently receiving more attention.

#### INNOVATIVE CONCEPTS

Innovative concepts (fig. 2(e)) are a relatively small part of the program. The thrust of the innovative program is to be sure that we don't miss any ideas. It may be analogous to the days of the piston engine for aircraft applications. We don't want to overlook something big like the jet engine.

Many studies are being conducted on augmentation devices such as the Coands diffuser shown in figure 16. Other types of augmentation, some a little more noval looking, are being examined. Figure 17 shows another augmentation device. If the diffuser of figure 16 had the shell cut away and just little pieces were left at the tips of the rotor blades, the configuration of figure 17 would be obtained. This study was done by AeroVironment on what they call a dynamic inducer. The designs mentioned here are not the entire effort. The intent is to illustrate the kinds of things that are included in the innovative program.

#### SYSTEM DEVELOPMENT

The end product of the program of course is systems development. Let's start with the small machines (fig. 2(f)), where small machines are defined as those with less than a 100 kW power output. There is a test center now in operation at the Rocky Flats facility of the Department of Energy in Colorado. Figure 18 shows a number of machines that are currently being tested at the Rocky Flats facility. Although they are now shown in the figure, there are a number of small vertical axis machines under development. Most of those shown are very small machines, but there are now development programs leading to larger machines. Figure 19 is another example of one of the commercial machines that is currently being tested at Rocky Flats. This evaluation is being done in an effort to help manufacturers to understand the capabilities of their machines and also to determine the performance of various configurations.

Figure 20 illustrates the three approaches that are being taken to develop highly reliable 1- to 2-kW size machines. There are two propeller type machines and a vertical axis machine. Eight-kilowatt size machines are shown in figure 21. There are four parallel developments in this category; again, a mixture of vertical axis and horizontal axis machines. In the 40-kilowatt size, as shown in figure 22, there are two parallel developments with one vertical and one horizontal axis machine. There are some advanced development programs about to be started in other sizes, specifically for 15 and 4 kW.



Let's now consider the machines in the intermediate and large size range. Intermediate size machines are arbitrarily defined as those with capacities larger than a 100 kW and yet smaller than 500 kW. Figure 2(g) lists the intermediate and large machines.

A machine that falls into the lower category is the MOD-0A. The installation in Culebra, Puerto Rico, was shown in figure 7, and the installation in Clayton, New Mexico, is shown in figure 23. The MOD-0A machine will be discussed extensively during the conference. The 2-mW MOD-1 machine, which is being installed in North Carolina, is shown in figure 24. For the largest, figure 25 shows an artist's concept of the  $2\frac{1}{2}$  mW MOD-2. This machine is now in the detailed design phase, and hopefully the machine will be running in the fall of 1980.

As far as advanced systems are concerned, we have recently received approval to initiate two new advanced systems. One is in the intermediate size range and the other is in the large multi-megawatt size range. This resulted after much reevaluation of the Wind Energy Program with regard to timing, the applications, and the markets for these machines. An RFP should be issued for the first of those machines later this fiscal year. The requirements of those machines, in particular the large one, merit some discussion.

A chart showing the cost trends of wind turbines is shown in figure 26. Three categories of costs are indicated in the upper righthand corner. The clear band represents preproduction costs. That refers to units purchased in groups of ones and twos, as opposed to the second slashed band below, which illustrates costs associated with limited production. This category represents a production of up to 25 machines or groups of 25 which still constitutes limited production. The dotted band represents what is considered the mature product projection (units of 100's).

On the left axis of figure 26 (note that the axis is broken with a changed scale), the first generation machines are the MOD-0's, MOD-0A's and the MOD-1. The costs for these very early prototypes are 20 to 30 cents a kW hour. Even if they were produced in large quantities and were located in very high wind sites, (like 16 mph), the cost would be reduced to only around 6 or 7 cents a kilowatt hour.

The second generation machines look a little different. The MOD-2, hopefully, will be able to attain a useful market for reasonable site wind speeds (down to 14 mi an hr) in the early 80's, to the point where it actually would compete with other fossil fuel-generating sources in areas where fuel costs are high. However, to really achieve broad and significant market, it is necessary that advanced systems be built that have a significant improvement over that which can be achieved with MOD-2.

Our goals are to produce machines at a cost of energy from 2 to 3 cents a kW hour. Our present program in the large machines is to have one more round of advanced system developments. By that is meant that there will be, budget permitting, parallel contracts to develop what are envisioned as the last generation of advanced machines. Invariably, there will be product improvement

programs and similar activities, but it is felt that the goal can be achieved in this round. The last set of bars to the right in figure 26 may be unnecessary.

For the intermediate sizes, the cost figures are somewhat less tight. The exact numbers are still being determined. We hope to achieve a useful market range in the intermediate size machines, but we still plan to have an additional cycle of advanced machines beyond the one that will be coming out this year.

How these goals and plans are going to be accomplished is really the topic of this meeting. Elements of the cost reduction activities are outlined in figure 27. We have asked NASA to supply a form of shopping list of the program activities that they feel will bring us from the current  $3\frac{1}{2}$  cents a kW hour down to  $2\frac{1}{2}$  cents a kW hour. It is strongly felt that weight and cost budgets can, in fact, be controlled to the point where a machine can be produced in that range. However, much technology work is needed to reach that cost level. Consequently, these programs will be structured to allow a significant amount of time at the start of the development programs to examine alternative ideas to achieve machine configurations that can attain those exacting goals.

The present workshop should be a fertile ground for ideas to suggest test programs that can investigate some of these areas. Hopefully, it will be an opportunity for people to interchange ideas and promote the task of developing required advanced systems.

#### PROGRAM RESOURCES

The budget for the Wind Program has seen considerable growth. In fiscal years 1973 and 1974 combined, it was about \$2 million. Last fiscal year, it grew to about \$36 million. The budget for this current year is about \$61 million. Figure 28 compares the 1978 and 1979 budgets. Although the value for next year is not yet known, it appears to be in excess of \$67 million. Numbers as high as \$100 million are heard. The Congressional hearings are going on currently, so the fiscal year 1980 budget is not known. However, a significant growth in that area is foreseen.

The other significant resources that we have to discuss is people. In addition to the expected two new slots in Washington, a new DOE Area Office will be set up to handle those aspects of the program that relate to small machine development, wind characteristics, and vertical axis machines. That office in Albuquerque, which will be set up by George Tennyson, is expected to be operational in July of this year.

#### SUMMARY

The Wind Energy Program has had significant growths, technically, organizationally, and budgetwise. However, a very significant challenge remains. The cost goals that have been established for advanced systems are tough. Very



aggressive technical development programs will have to be mounted to achieve them. As can be seen from the organization of this conference, a number of unknowns have been recognized that will require addressing. However, it is the unknown unknowns that are worrisome. Hopefully, this conference will provide an opportunity for people to surface potential problem areas that should be investigated.

That basically is the challenge. Our strengths are recognized and acknowledged, but let's try to identify our weaknesses. Hopefully, in so doing, our goals will be reached.

#### REFERENCE

Wind Energy Systems Program Summary. U.S. Department of Energy, Office of Solar, Geothermal, Electric and Storage Systems. DOE/ET-00093, December 1978.

#### DISCUSSION

- Q. Is work continuing on offshore site selection?
- A. With regard to site selection, proposals are invited from any site area. There have been no proposals for offshore sites. There is a study conducted by Westinghouse that showed very difficult technical problems and high costs with placing machines offshore. This study is just about complete, and it should be published in the near future. There is no question that there are good resources offshore, but the technical problems and costs are of major consideration.
- Q. You have presented a very good program which concentrates on Government efforts. Since the stated objective is to put the Government out of business, what efforts are being made to relate to the private activities that are going on in wind turbine development, such as Schackle and others?
- A. The Federal Wind Program was summarized, since that was my assignment. However, our feeling about the private effort is that if there is anything we can do to help and encourage that effort, we will respond. It is excellent that private ventures are starting. It is a sign that a healthy market may be developing.

Federally sponsored R & D activities are open to anyone. We issue a RFP for advanced systems, and anyone can bid. Private companies that chose not to bid on federal contracts, for reasons which are understandable, can shift from the private sector into the government R & D at any time. Also, the reports that are published and the technology that is developed under government funding are available to private entrepreneurs. We encourage the use of such information, and when requests are received, we generally will supply bibliographies and any knowledge that is available.

## OVERALL APPROACH

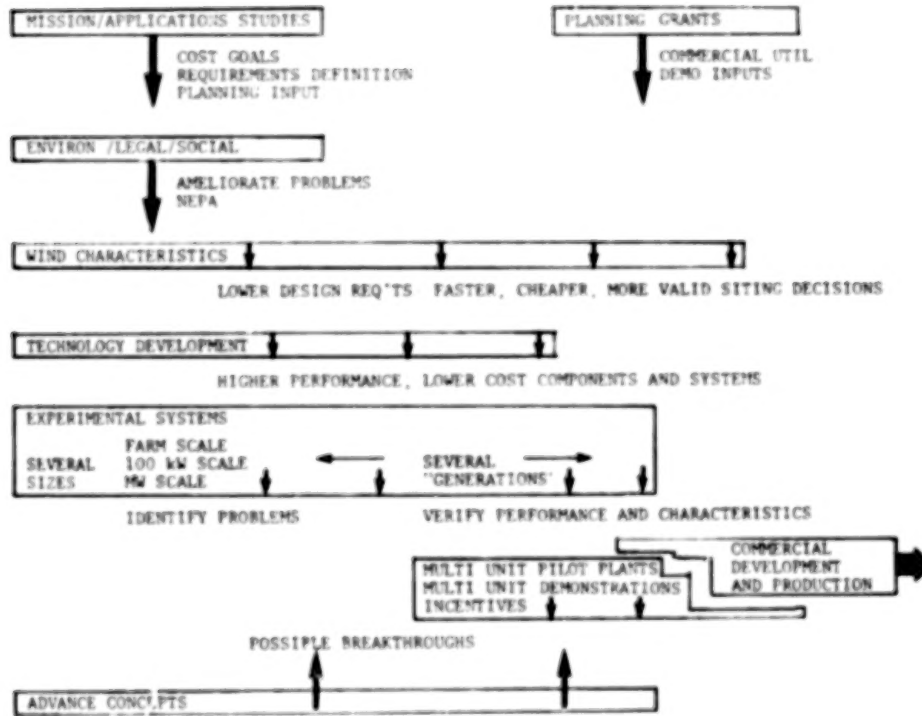


Figure 1

## MAJOR PROGRAM ELEMENTS

- |  |   |
|--|---|
| <p>(a) MISSION AND APPLICATIONS ANALYSIS</p> <ul style="list-style-type: none"> <li>* Mission and Market Studies</li> <li>* Utility and Specific Applications</li> <li>* Candidate Sites</li> </ul> <p>(b) LEGAL SOCIAL AND ENVIRONMENTAL ISSUES</p> <ul style="list-style-type: none"> <li>* Machine Aesthetics</li> <li>* Siting and Land Use</li> <li>* Wind Rights</li> <li>* Energy Cost and Rates</li> <li>* Codes and Standards</li> <li>* Environmental Impacts</li> </ul> <p>(c) WIND CHARACTERISTICS</p> <ul style="list-style-type: none"> <li>* Resource Assessments</li> <li>* Area Survey Techniques</li> <li>* Siting Handbooks</li> <li>* Forecasting and Operation</li> </ul> | <p>(d) TECHNOLOGY DEVELOPMENT</p> <ul style="list-style-type: none"> <li>* Experimental Machine</li> <li>* Component Development</li> <li>* Analytical Model: Aerodynamic; Structural Dynamic; Power Grid Stability</li> </ul> <p>(e) INNOVATIVE CONCEPTS</p> <ul style="list-style-type: none"> <li>* Augmentation</li> <li>* Vertical Axis</li> <li>* Unique Approaches</li> </ul> <p>(f) SMALL SYSTEMS DEVELOPMENT</p> <ul style="list-style-type: none"> <li>* Test Center</li> <li>* Commercial Machine Tests</li> <li>* New Systems Development</li> </ul> <p>(g) LARGE SYSTEMS DEVELOPMENT</p> <ul style="list-style-type: none"> <li>* MOD 0A      200 kW machines</li> <li>* MOD 1        2 MW machine</li> <li>* MOD 2        2.5 MW machine</li> </ul> |
|--|---|

Figure 2

**WIND SPEED CONTOURS IN THE  
NEGEA CAPE COD SERVICE REGION**  
(reference height = 10 meters)

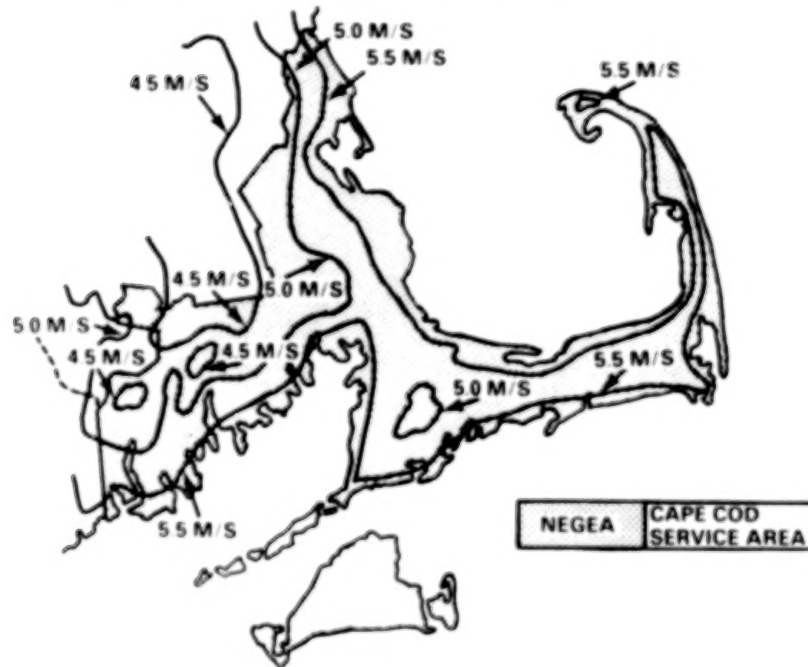


Figure 3

**FARM WINDMILL (U.S.D.A.)**



Figure 4

# WTG CANDIDATE AND INSTALLATION SITES



Figure 5

## ARTIST CONCEPTION OF MOD-2 TURBINES ON A RIDGE

(Machines spaced 10 rotor diameters apart)



Figure 6

MOD-0A AT CULEBRA, PUERTO RICO

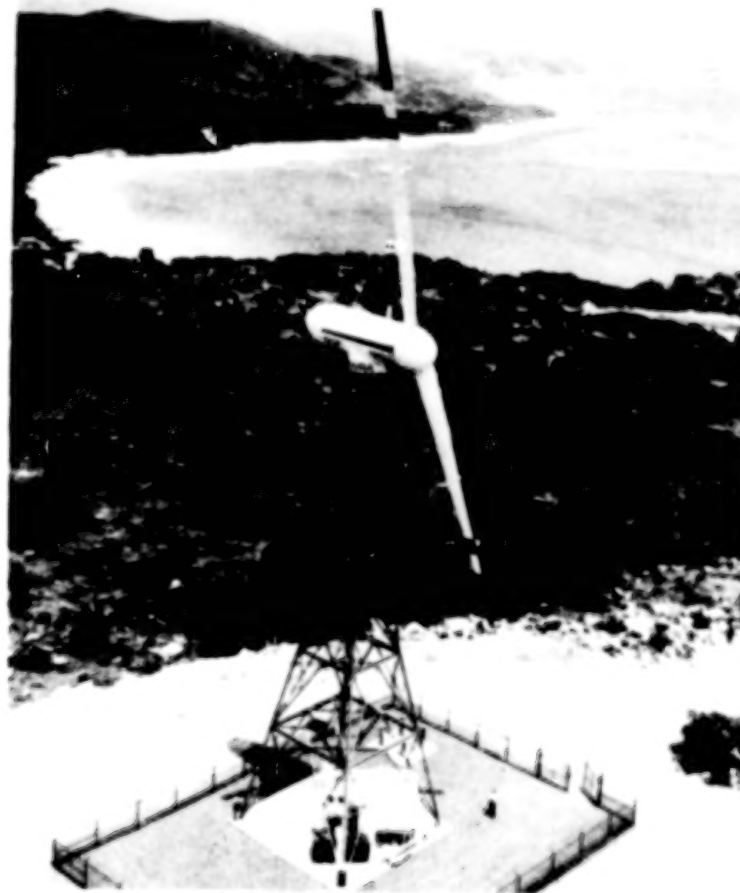


Figure 7

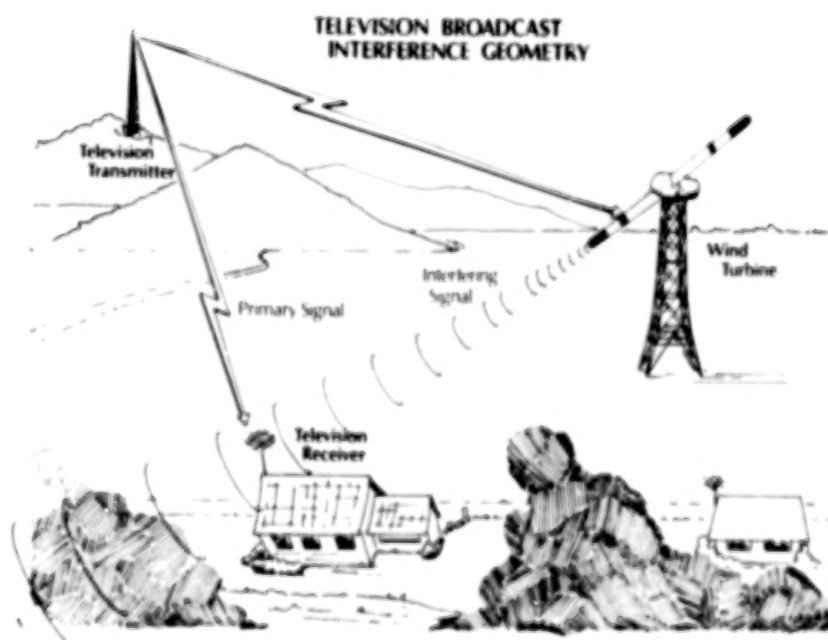


Figure 8

# ANNUAL AVERAGE WIND POWER (WATTS/M<sup>2</sup>) AT 50 M

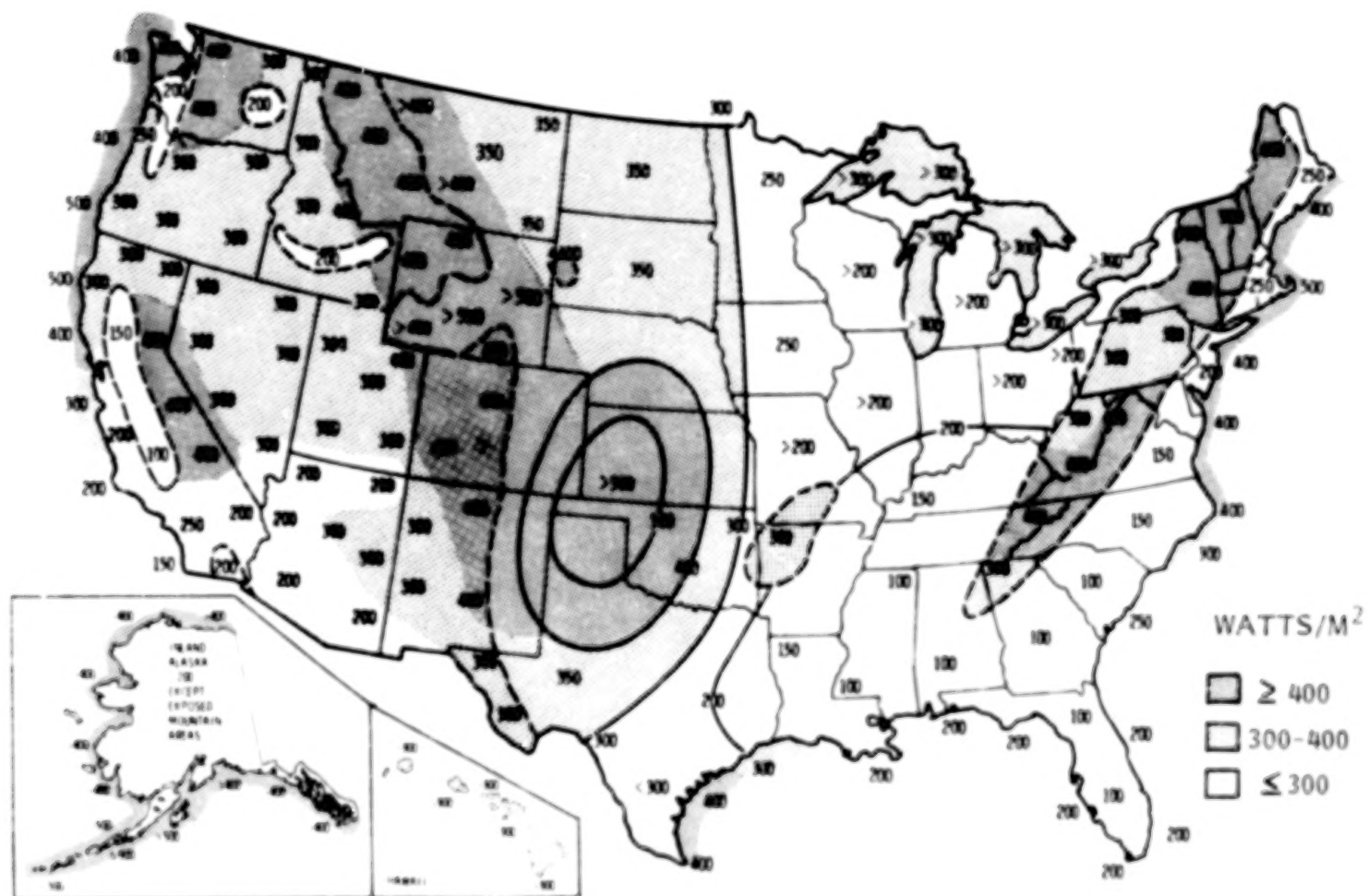


Figure 9



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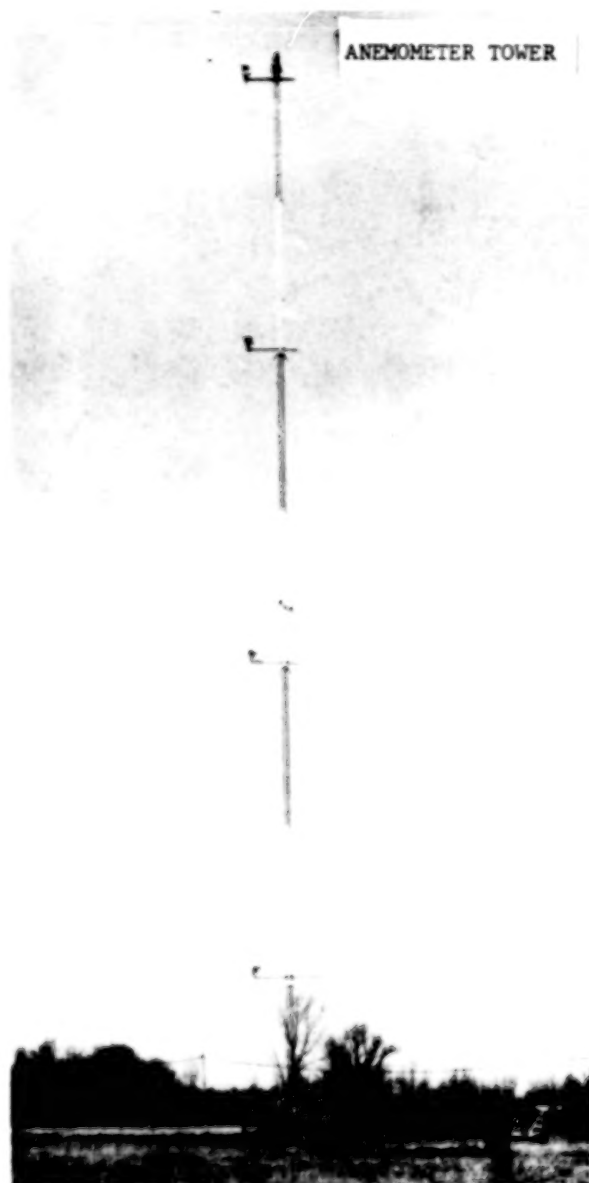


Figure 11

MOD-0 EXPERIMENTAL WIND TURBINE  
Plum Brook, Ohio

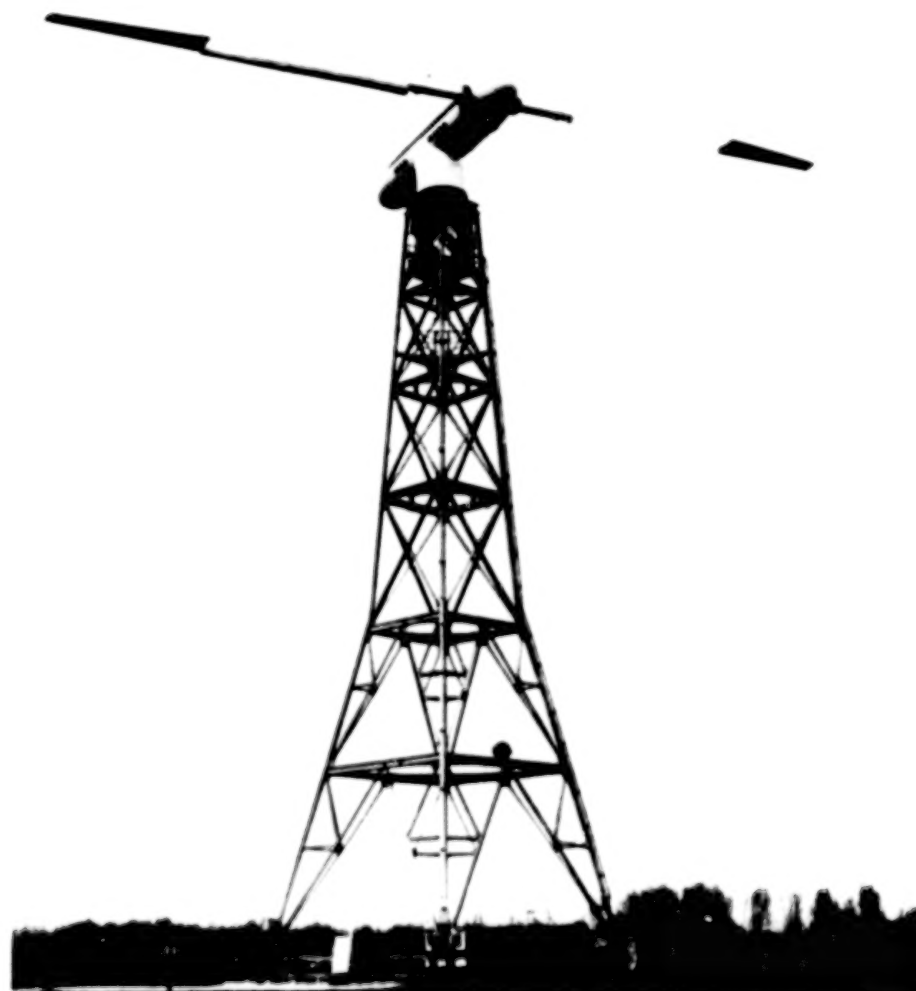


Figure 12

150-FT FIBERGLASS BLADE

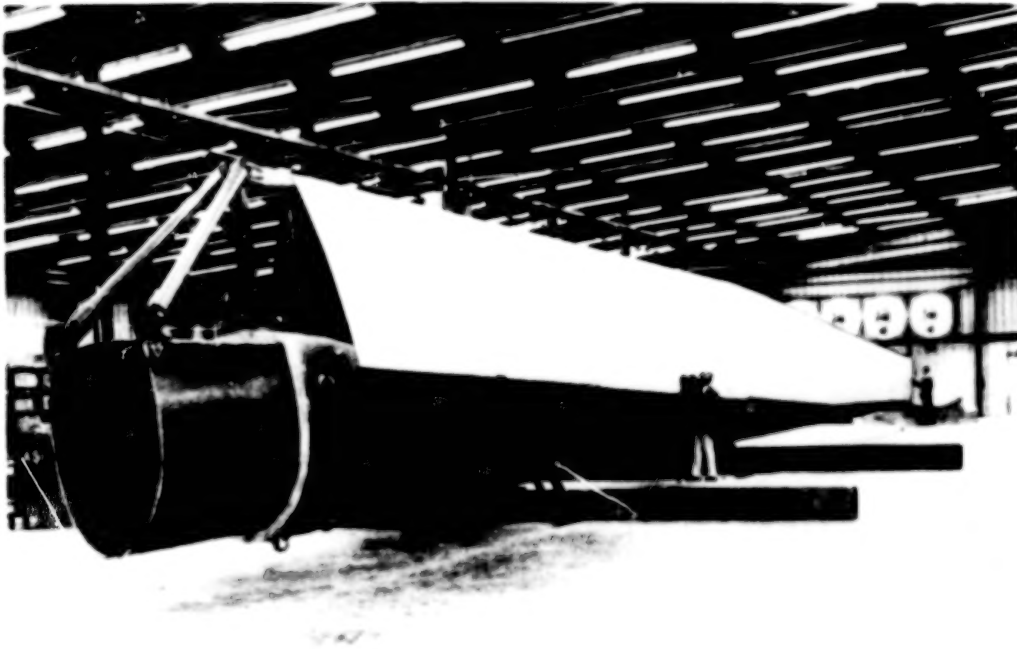
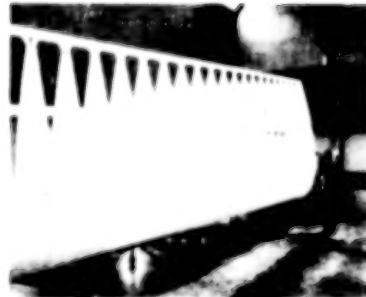
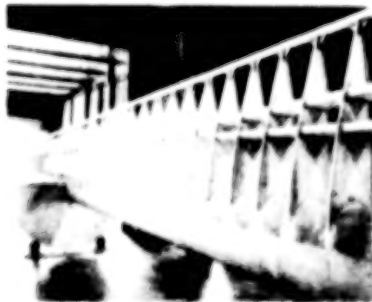


Figure 13

NASA  
TM X-2001

150-FT FIBERGLASS BLADE



FABRICATION



LOAD APPLICATION



INSTALLATION IN MODEL

Figure 14

SARRIEUS VERTICAL AXIS WIND TURBINE

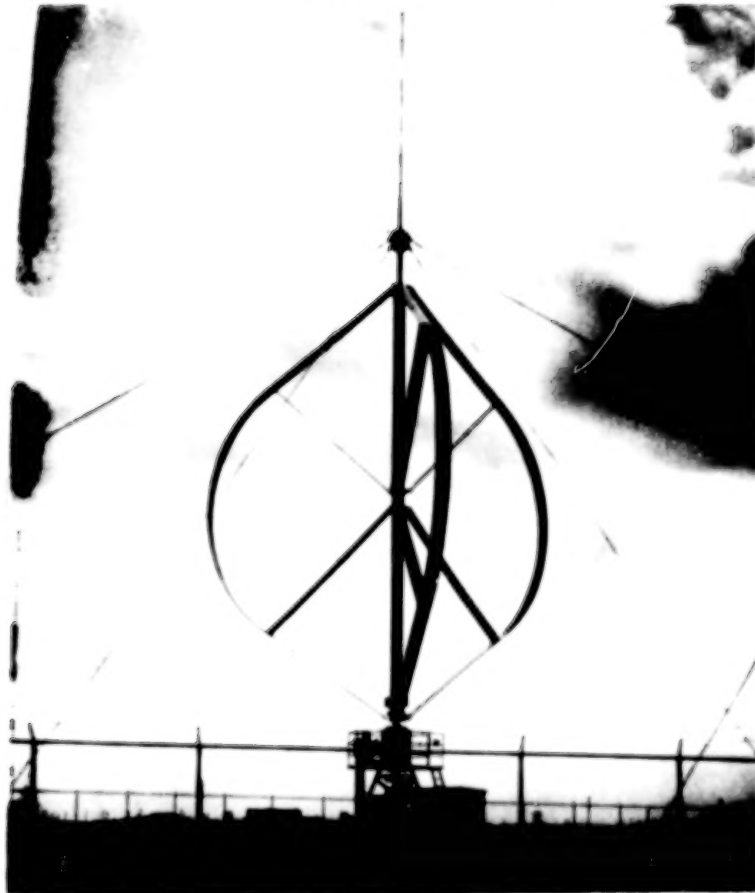


Figure 15

COARDA DIFFUSER CONCEPT

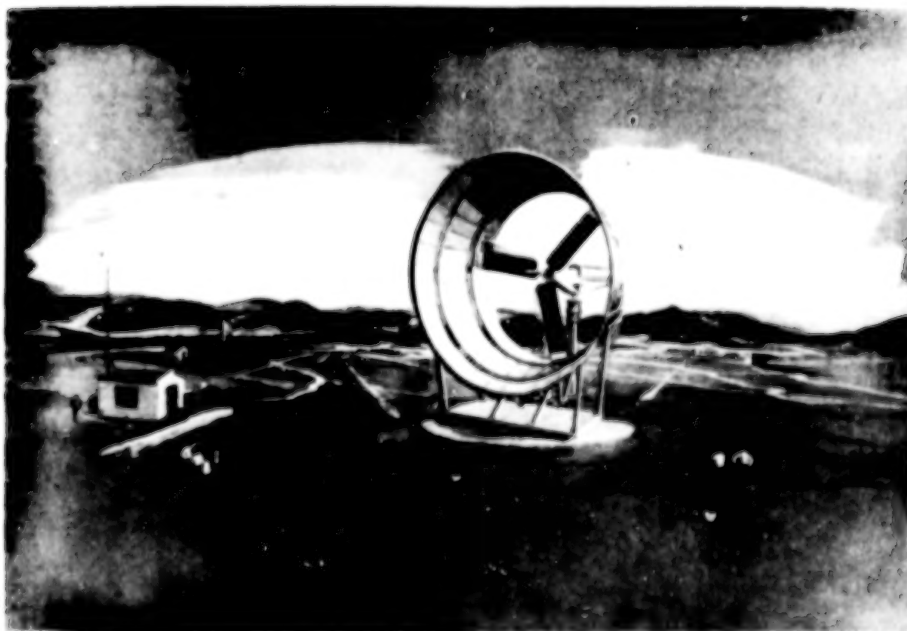


Figure 16

DYNAMIC INDUCER CONCEPT



Figure 17

ROCKY FLATS TEST FACILITY



Figure 18



Figure 19

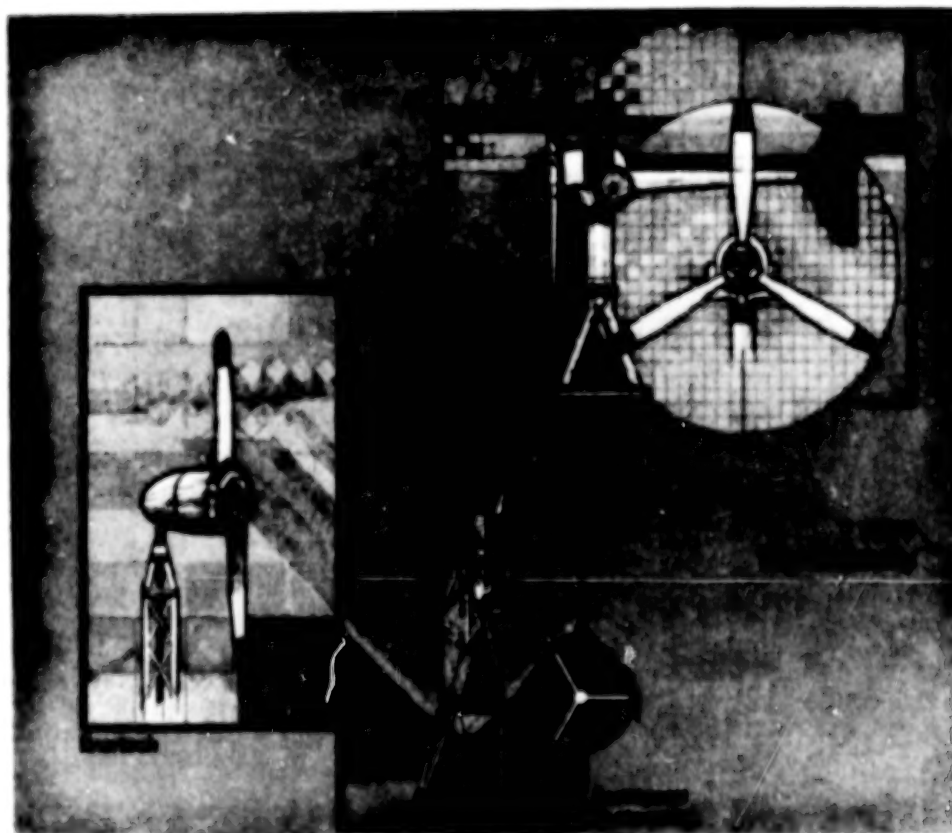


Figure 20

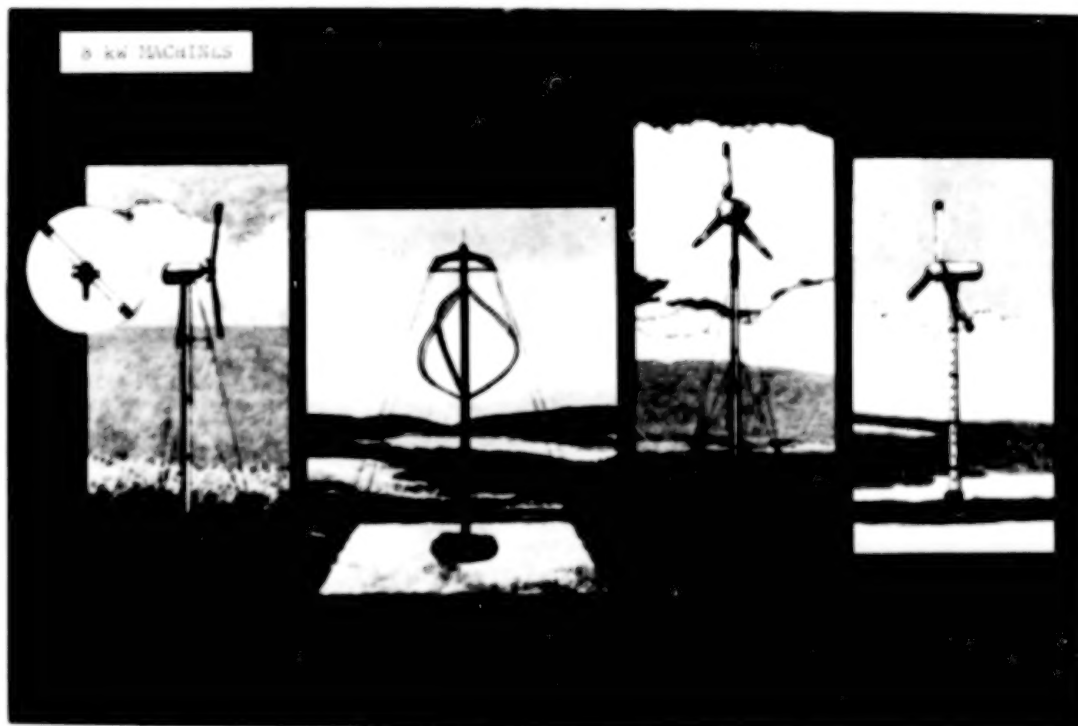
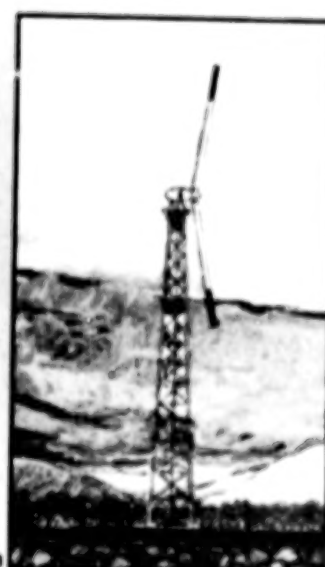


Figure 21



McDonnell  
Aircraft Company

40 kW



Kaman

Figure 22



MOD-0A

CLAYTON, NEW MEXICO



Figure 23

DOE/NASA 2000kW EXPERIMENTAL WIND TURBINE

Howard's Knob, Boone, North Carolina

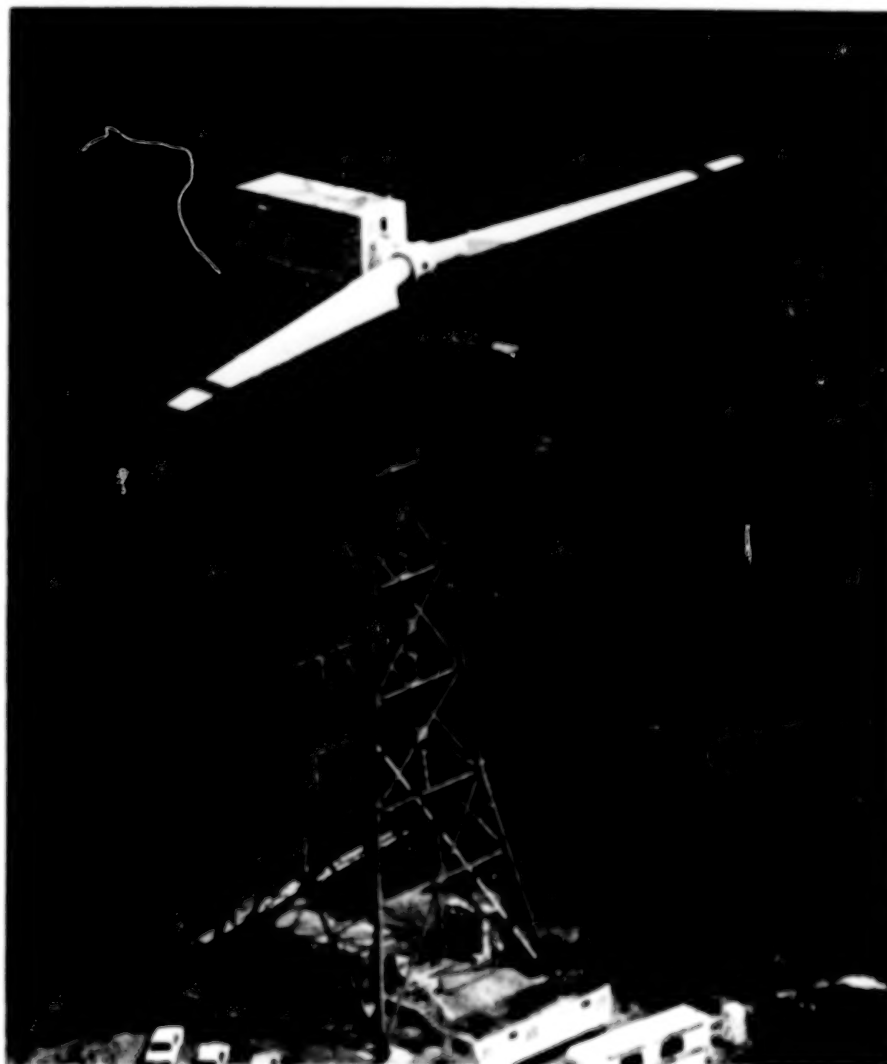


Figure 24

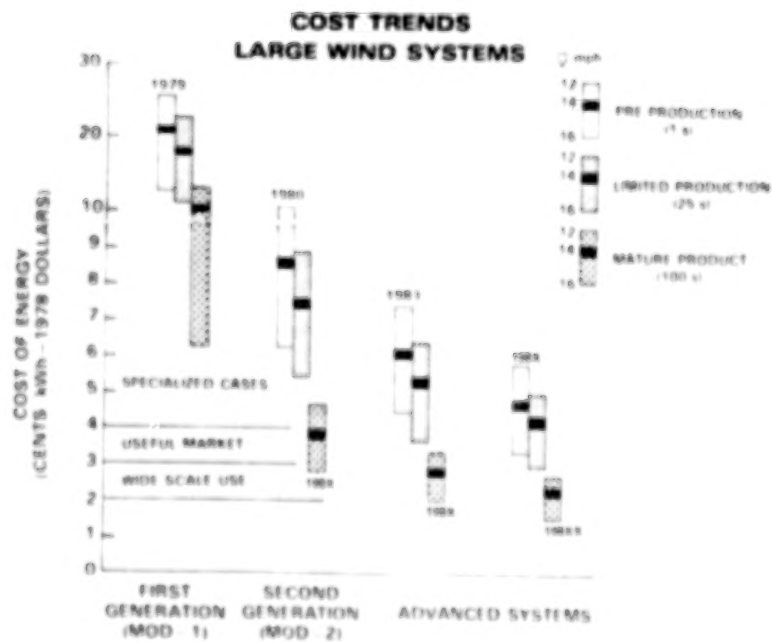


Figure 26

### TECHNOLOGY TO REDUCE COE

- MATURE MACHINE COE FROM 3.5¢/kWh TO 2.5¢/kWh
- DETAILED WEIGHT AND COST BUDGETS AND SENSITIVITIES DEVELOPED
- RELATED SUPPORTING TECHNOLOGY PLAN
- TECHNOLOGY AREAS
  - FLAP / SPOILER
  - FIBER GLASS / FABRICATION TECHNIQUES
  - MULTI-SPEED RPM
  - TIPS AND AIRFOILS
  - PASSIVE YAW
  - DAMPING ALLOWING COMPACTION
  - GEARBOX INTEGRATION
  - SELF-ERECTING
  - LOWER LOADS AND FREQUENCIES
  - CONTROL ALGORITHMS
  - SYSTEM CONFIGURATION OPTIMIZATION

Figure 27

### Wind Energy Conversion Systems FY 1979 Budget (\$ in Millions)

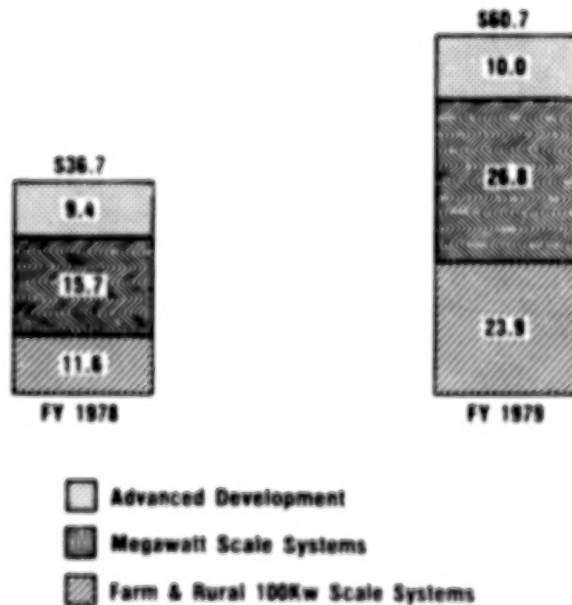


Figure 28

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## DESIGN EVOLUTION OF LARGE WIND TURBINE GENERATORS

David A. Spera  
NASA Lewis Research Center  
Cleveland, Ohio

### SUMMARY

The design of large wind turbines of the horizontal-axis type has evolved rapidly during the past five years (fig. 1). Major changes have taken place in the structural and mechanical features of second generation wind turbines like the 2.5 MW Mod-2 (fig. 2), compared with first generation machines like the 200 kW Mod-OA (fig. 3) and 2.0 MW Mod 1. These changes have reduced the projected cost of electricity produced by second generation wind turbines to one-half that of first generation systems. Furthermore, wind machines like the Mod-2 have been designed to take advantage of the economies of mass production, so electricity generation costs are expected to eventually be cut in half again. Thus, during the past five years the goals of economy and reliability have led to a significant evolution in the basic design - both external and internal - of large wind turbine systems.

To show the scope and nature of recent changes in wind turbine designs, developments of three types are described: (1) system configuration developments; (2) computer code developments; and (3) blade technology developments. Developments in system configuration are shown by direct comparison of Mod-2 components (fig. 4) with equivalent elements in the earlier Mod-OA system (fig. 5). Significant economy has been achieved in blades by changing from lightweight but expensive aluminum aircraft construction to heavier but cheaper welded steel fabrication. As a result, rotor costs which were disproportionately high in the Mod-OA system now account for less than 25 percent of the Mod-2 cost of electricity (fig. 6). In addition, heavy and rigid elements like the Mod-OA tower, hub, and drive-train bedplate have evolved into lighter, more flexible, and more economical components in the Mod-2 machine.

Computer code development (fig. 7) has closely paralleled and supported configuration development. Special-purpose computer codes are now available for predicting the aerodynamic performance and structural dynamic behavior of large horizontal-axis wind turbines. Both proprietary and non-proprietary codes (with development and verification coordinated by LeRC) are listed in figure 8. Sources for detailed information on these codes are given in figure 9. Application of the newly-developed MOSTAS code is illustrated by comparing calculated dynamic blade loads with loads measured on the 100 kW Mod-0 test turbine (figs. 10 and 11).

Blade costs are one of the most important factors in determining the cost of generating electricity by wind power. Therefore, a wide variety of developments in blade design have occurred in the past five years, with the goal of reducing both initial cost and maintenance. Seven different blade designs are described (fig. 12) to illustrate the evolution which has taken place.

The trend is toward the use of materials and manufacturing processes that produce blades which are lower in relative cost but higher in relative weight, compared to the complete wind turbine.

While design improvements in second-generation wind turbine generators have significantly reduced the projected cost of electricity, further improvements are expected in the near future. The design of large wind turbines will continue to evolve, based on new technology and operating experience with present machines (fig. 13).

#### DISCUSSION

Q. What is the percentage cost associated with the Mod-1 blades and the Mod-2 blades?

A. The cost of the two Mod-1 blades is about 34 percent of the installed cost of the whole system. For the Mod-2 machine the blades represent about 25 percent of the capital investment. Now the bar chart I showed was based on the cost of electricity, which includes not only capital investment but also operation and maintenance costs. So there will be some small differences in the percentages.

The breakdown of weights and the approximate cost percentages will be given in some of the later presentations. There is hesitation, sometimes, on cost breakdowns because all the machines are not directly comparable. On the Mod-1 there are blades that are very expensive. On the Mod-2 the hub is an integral part of the blades, so we speak of rotors. Many times we try to compare Mod-1, Mod-0A and Mod-2 and it becomes a real problem. We would be happy to give you the actual dollar values behind the bar charts.

Q. At the time the requests for proposal went out for Mod-2, had DOE made the decision for a soft tower, or did the soft tower happen to win out?

A. The latter is the case. The soft tower was proposed by the Boeing Engineering and Construction Company which was the winner of the Mod-2 contract.

Q. As to all of these features that you outlined that contributed to the weight reduction, were they all fixed at the time the decision was made to go that way, or were some of them developed as the design process went along?

A. Some were developed during the conceptual and preliminary design processes. At the beginning of the Mod-2 effort, there were extensive trade studies conducted by Boeing: soft tower versus hard; two blades versus three blades; upwind rotor versus downwind. What you see here are the results of those studies.

## DOE/NASA HORIZONTAL-AXIS WIND TURBINE GENERATORS

DESIGN		RATED POWER, KW	ROTOR CHARACTERISTICS				TOWER TYPE	COE, * UNIT 2 ¢/KWH
MOD	YEAR		DIAM, FT	HUB TYPE	LOCATION	WEIGHT, % TOTAL		
FIRST GENERATION								
0	'74	100	125	RIGID	DOWNWIND	4.7	STIFF	---
0A	'76	200	125	RIGID	DOWNWIND	5.1	STIFF	37
1	'78	2000	200	RIGID	DOWNWIND	5.6	STIFF	17
SECOND GENERATION								
2	'79	2500	300	TEETER	UPWIND	26.7	SOFT	8

\*14 MPH SITE

Figure 1

### Wind Turbine Configuration MOD-2

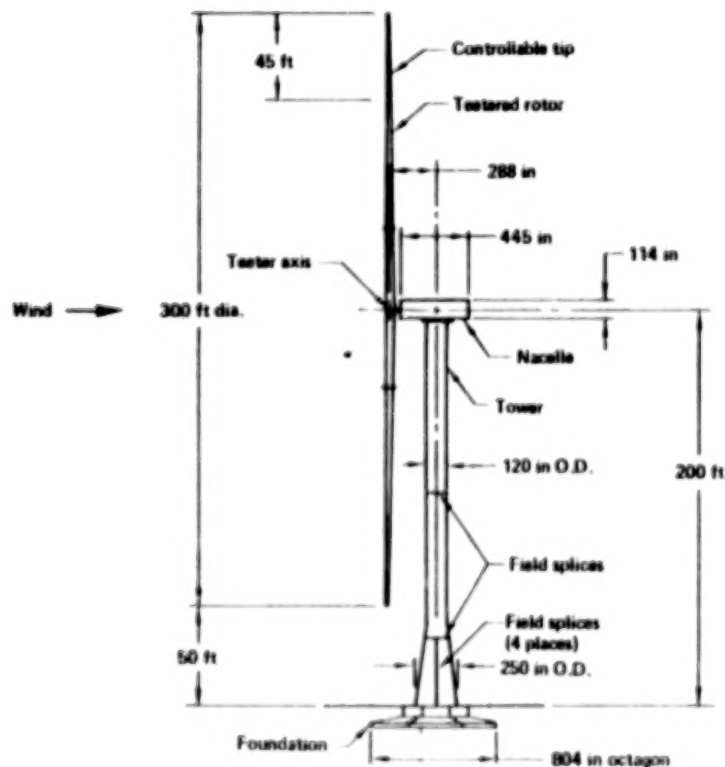


Figure 2

# MOD-0A WIND TURBINE



Figure 3

## 2500 KW MOD-2 WIND TURBINE

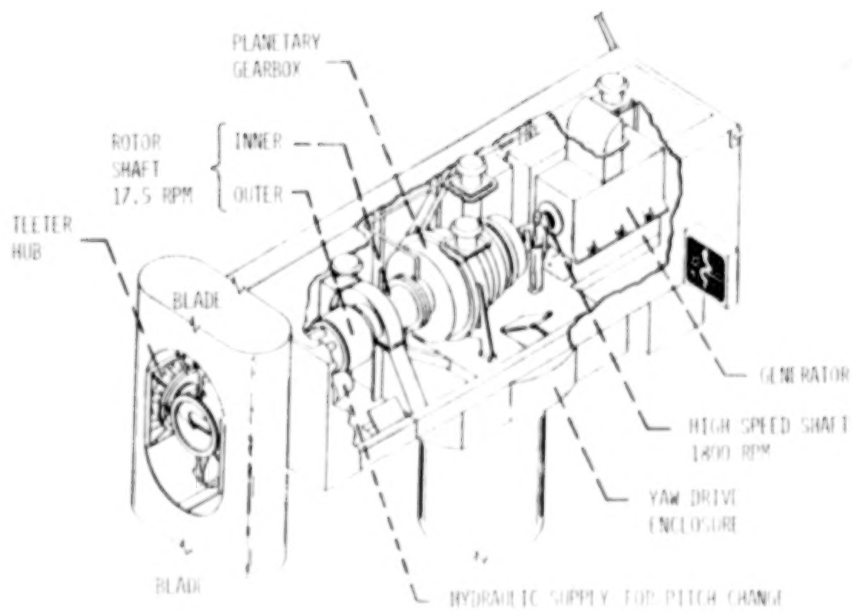


Figure 4



## 200 KW WIND TURBINE

MOD-0A

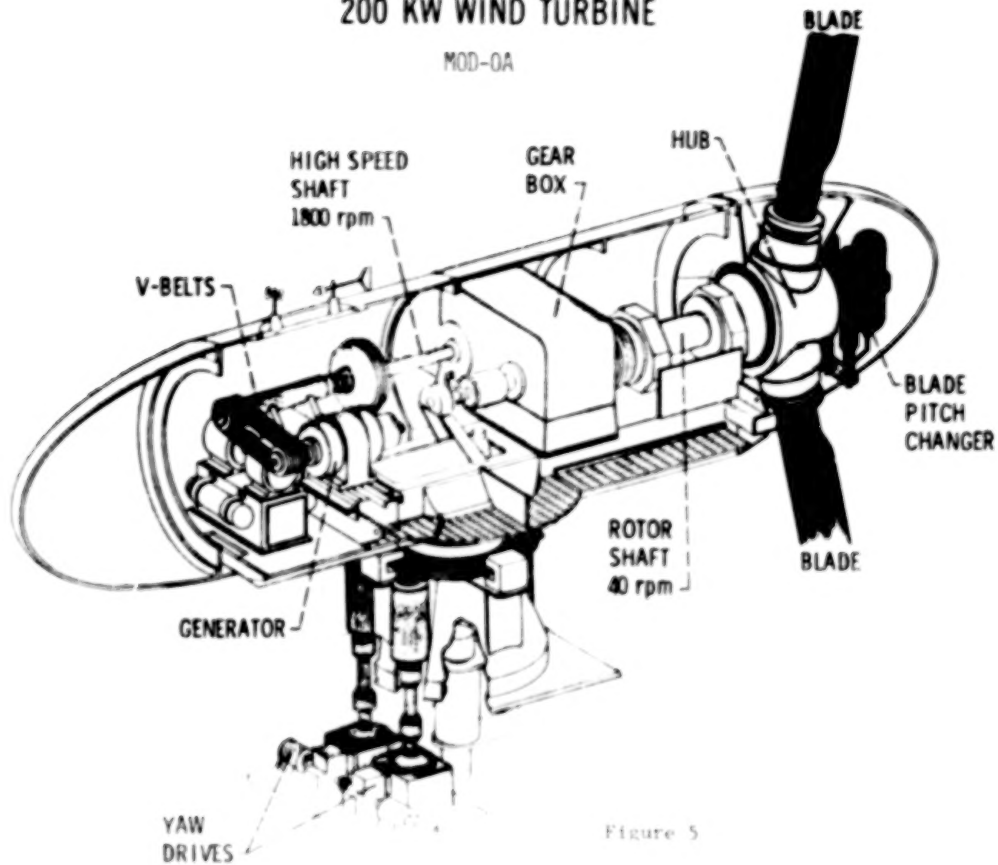


Figure 5

### CONTRIBUTION OF DESIGN ELEMENTS TO COST-OF-ELECTRICITY

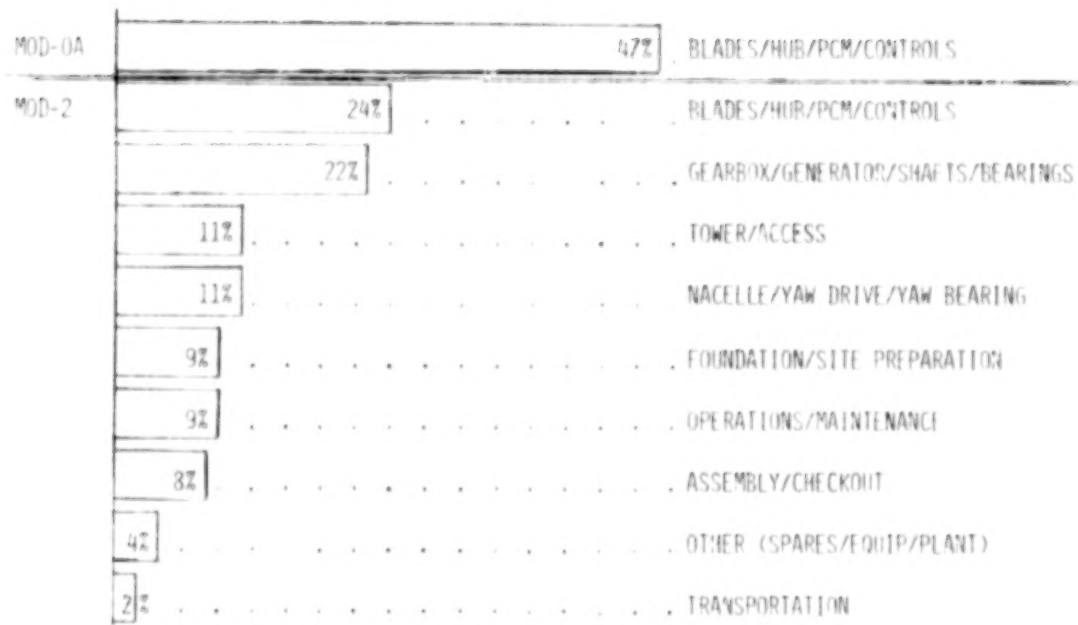


Figure 6

## WTG CHARACTERISTICS REQUIRE SOPHISTICATED ANALYSIS CODES

- LARGE FLEXIBLE ROTATING AIRFOILS, SUBJECT TO AEROELASTIC LOADS, COUPLED LOADS, AND DYNAMIC INSTABILITY
- LONG-LIFE STRUCTURES (UP TO 30 YR), SUBJECT TO FATIGUE
- ALL-WEATHER MACHINE, SUBJECT TO HIGH WINDS, SNOW, ICE, RAIN, DUST, TEMP EXTREMES, VANDALISM
- AIR LOADS ARE TRANSIENT, CYCLIC, AND STOCHASTIC
- EFFICIENT PERFORMANCE REQUIRED, SUBJECT TO CUT-IN, CUT-OUT, POWER CONTROL, YAW CONTROL, AND WIND PROBABILITY
- AUTOMATIC, UNATTENDED, REMOTE, FAILSAFE OPERATION REQUIRED, WITH LOW OPERATIONS AND MAINTENANCE BUDGET

Figure 7

## STRUCTURAL DYNAMICS COMPUTER CODES FOR HORIZONTAL AXIS WIND TURBINES

### 1. NON-PROPRIETARY CODES

- AUTHOR: PARAGON PACIFIC, INC.
- MOSTAB-WT SINGLE BLADE, 1 DEGREE OF FREEDOM (DOF)
- MOSTAB-WTE LERC EMPIRICAL ADDITIONS
- MOSTAB-HFW 4 DOF ROTOR, PLUS TEETERING
- MOSTAS COMPLETE WTG SYSTEM; MOD-2 APPLICATION BY BEC

### 2. PROPRIETARY WTG SYSTEM CODES

- REXOR-WT LOCKHEED-CALIFORNIA
- GETSS GE SPACE DIVISION
- F-762 UNITED TECHNOLOGY RES. CENTER

### 3. VERIFICATION REQUIRED OF ALL CODES

- MOD-0 LOAD DATA
- MOD-2 1/20 SCALE WIND TUNNEL MODEL DATA

### 4. WEST WTG SIMULATOR

- HYBRID ANALOG/DIGITAL COMPUTER
- USES MOSTAS SOFTWARE
- SPEED INCREASE BY FACTOR OF 100

Figure 8

# AVAILABLE STRUCTURAL-DYNAMIC CODES

CODE	SOURCE
MODTAB-WT	Mr. Barry Holchin Mechanics Research Incorporated 9841 Airport Boulevard Los Angeles, CA 90045
MODTAB-WTE	Dr. David A. Spera NASA-Lewis 49-6 21000 Brookpark Road Cleveland, OH 44135
MODTAB-HFW	Mr. John A. Hoffman Paragon Pacific Incorporated 1601 E. El Segundo Boulevard
GETTS	Mr. Clyde Stahle General Electric Space Division Box 8661 Philadelphia, PA 19101
F-762	Dr. Richard Bielawa United Technologies Research Center East Hartford, CT 06108
MODTAS	Mr. John A. Hoffman Paragon Pacific Incorporated 1601 E. El Segundo Boulevard El Segundo, CA 90245
PEROR-WT	Mr. Robert E. Donham Dept 75-21, Bldg. 360, Plant B-6 Burbank, CA 91520

## Reference

"Comparison of Computer Codes for Calculating Dynamic Loads in Wind Turbines.  
by D. Spera. NASA TM-77773 and DOE/NASA/1028/78-16, 1978.

Figure 9

## TYPICAL EDGEWISE MOMENT LOAD MOD-0 WIND TURBINE BLADE SHANK

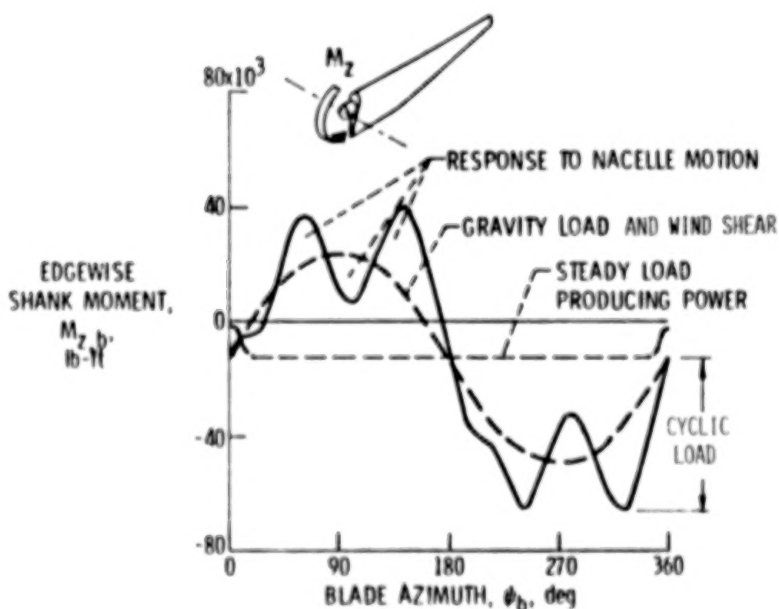


Figure 10

# MOD-0 BLADE LOADS VS. YAW DRIVE STIFFNESS

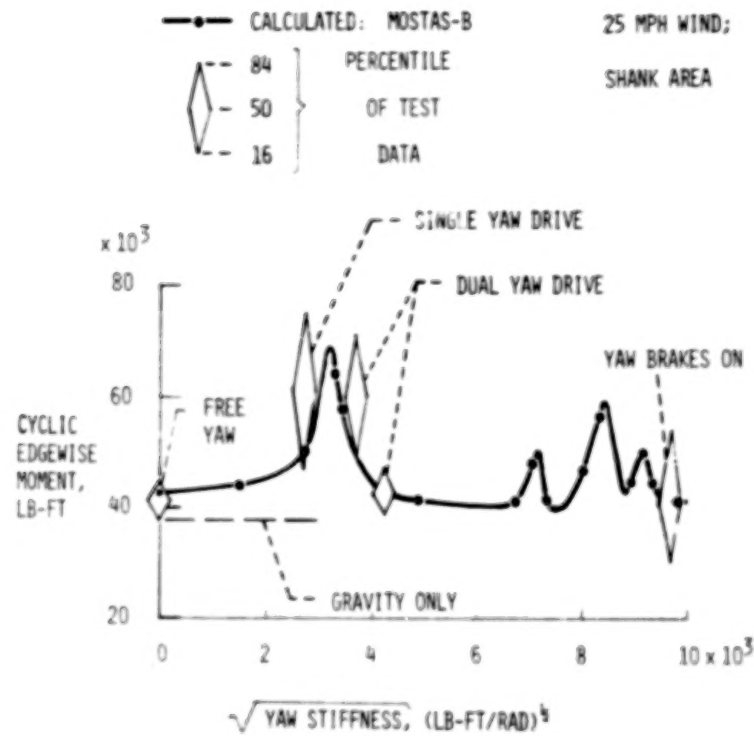


Figure 11

## WTG BLADE CONFIGURATIONS

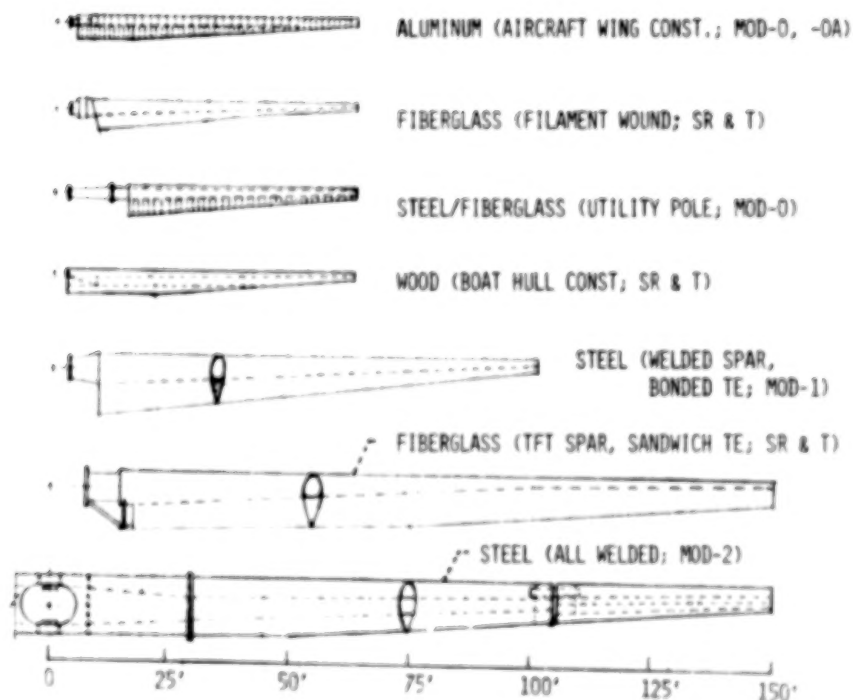


Figure 12

## DESIGN OF LARGE, HORIZONTAL AXIS WIND TURBINE GENERATORS

1. WTG DESIGN REQUIRES SOPHISTICATED TECHNOLOGY BACKED BY SPECIALIZED ANALYTICAL TOOLS.
2. THESE TOOLS ARE AVAILABLE NOW, BUT THEY REQUIRE CONTINUOUS MAINTENANCE, VERIFICATION, AND UPGRADING.
3. DESIGN IMPROVEMENTS -- VALIDATED BY ANALYSIS AND MOD-0 TESTS -- HAVE:
  - REDUCED STRESSES IN MOD-0A WTG
  - REDUCED ROTOR COSTS IN MOD-2 WTG
  - CONTROLLED COSTS OF MOD-2 TOWER AND NACELLE
4. DESIGN REQUIREMENTS AND METHODS WILL CONTINUE TO EVOLVE, TO INCLUDE:
  - NEW MOD-0, -0A, AND -1 TEST DATA
  - MORE ANALOG SIMULATION AND GRAPHICS
  - MORE STATISTICAL DATA ON WIND LOADS
  - IMPROVEMENTS IN FATIGUE AND BUCKLING ANALYSES
  - DESIGN HANDBOOKS

Figure 13

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## THE GENERAL ELECTRIC MOD-1 WIND TURBINE GENERATOR PROGRAM

Richard H. Poor and R. B. Hobbs  
General Electric Co., Space Division  
Valley Forge, Pennsylvania

### INTRODUCTION

The MOD-1 WTG is the first megawatt class machine in the national wind program. The MOD-1 Program which started in September of 1976 has as its objectives the design, fabrication, installation and test of a megawatt class wind turbine generator (WTG) which generates utility grade electrical power. The program is nearing the final phase of installation and checkout. The blades are the only components remaining to be installed.

NASA-LeRC is managing the MOD-1 Program for DOE. General Electric's Space Division located in Valley Forge, PA., is the prime contractor, with several GE electrical equipment product departments supplying components ranging from switchgear to the synchronous generator.

### WTG SPECIFICATIONS AND REQUIREMENTS

The specifications and design requirements, as originally stated by NASA-LeRC were heavily influenced by the NASA MOD-0 design and operational experience, and as such, the designs have a high degree of similarity. Also, the MOD-1 technical specifications are quite restrictive and allowed virtually no flexibility in design concept except for a trade-off between a rigid and a teetered hub. The general design requirements and program objectives are shown in figure 1. The dominant requirement, which most influenced the design, was the utilization of state-of-the-art technology to minimize technical risk.

A summary of the technical specifications is contained in table I. You will note that the selection of a few of the parameters was optional. Also during the design cycle, a requirements assessment analysis was conducted, which lead to the modification of certain requirements. These items will be reviewed in the discussion on design drivers.

The design wind environment for the MOD-1 WTG is an 18 mph (mean) wind regime with a standard Velocity Duration Curve. The vertical velocity profile is defined by the relationship:

$$V = V_0 \left( \frac{H}{H_0} \right)^{0.167} \quad V_0 = \text{velocity at ref. height } H_0$$

The wind gust model is shown in figure 2.



The blade design load cases are listed in table II. The blade turning requirements were a first flapwise frequency  $\geq 2.15 P$  ( $P$  = nominal rotational frequency) and a first chordwise frequency  $\geq 4.15 P$ . The primary requirements of the pitch change mechanism were a maximum pitch rate of  $8^\circ/\text{sec}$  and a stiffness of  $20 \times 10^6$  inch-pound per radian.

The generator specification was 4160 V, 3 $\phi$ , Y-connected, synchronous, 1875 kVa at 60 Hz. Emergency power was to be provided by an auxiliary power unit in the control enclosure. Slip rings or loop cable were indicated for the power connection at the nacelle. Protection items include conventional switchgear as well as lightning protection. Electrical system stability was required for 5 to 1000 MW. The control system functional requirements include startup and synchronization, shutdown, and maintenance of electrical stability. Unattended operation is called for with manual operation from the WTG site. Remote monitoring and control by power dispatcher is also required. Finally, an engineering data acquisition system should be provided.

#### DESIGN DESCRIPTION

The MOD-1 WTG has a configuration which is depicted in the photo of a scaled model shown in figure 3. It has a 200-foot two-bladed downwind rotor that operates at a constant 35 rpm with its axis at an elevation of 140 feet. An outline drawing (fig. 4) defines the basic WTG dimensions.

The rotor drives a synchronous generator through a speed increaser gearbox. Synchronous speed and power are controlled by varying blade pitch as the wind speed varies. The tower is a 131-foot truss structure with a 48-foot base. A control enclosure and transformer are installed at ground elevation below the WTG within the envelope of the four-tower legs. The major elements of the WTG are briefly described as follows:

a. Rotor. - Two steel blades are attached to the hub barrel via three-row cylindrical roller bearings which permit the pitch angle of the blade to be varied 105 degrees from full feather to maximum power. Blade pitch is controlled by hydraulic actuators which provide a maximum pitch rate of 14 deg/sec. Figure 5 shows the pitch control block diagram.

The hub tailshaft provides the connection to the low-speed shaft and to the dual tapered roller main bearing, which supports the rotor and one end of the low speed shaft.

Each blade is tapered in planform and thickness as shown in figure 6. It utilizes a NACA 44XX series airfoil with thickness ratio varying from 33% at the root to 10% at the tip. The twist of  $11^\circ$  varies linearly from root to tip. The blades are mounted on the hub at a  $9^\circ$  cone angle to optimize stresses due to thrust and centrifugal forces.

The blade in final assembly is shown in figure 7. The major blade load carrying member, a hollow spar, is fabricated from A533 Grade B, class 2 high strength, low carbon steel. The trailing edge is fabricated from urethane foam sections with 301 stainless steel skins.

b. Drive train. - Figure 8 shows the drive train which consists of the low-speed shaft and couplings, a three-stage gearbox and the high speed shaft which drives the generator. The gearbox has parallel shafts. The high speed shaft incorporates a dry disk slip clutch for protection against torque overloads, and a disk brake that stops the rotor and holds it in the parked position. The gearbox lubrication system also provides oil to the rotor bearing and dissipates waste heat by means of a cooler suspended below the nacelle.

c. Power generation/control. - Figure 9 is a block-diagram of the power generation/control system. A GE synchronous AC generator is driven at 1800 rpm by the high-speed shaft. A shaft mounted, brushless exciter, controlled by a solid state regulator and power stabilizer provides voltage control. Generator output at 4160 volts is brought by cables and a slip ring at the yaw bearing down the tower to the control enclosure and then on to the utility interface via a 2000 kVA step-up transformer.

d. Nacelle structure. - A welded steel bedplate is the primary structure, supporting all equipment mounted on top of the tower and providing a load path between the rotor and yaw structure. Other equipment mounted on the bedplate includes the pitch control and yaw drive hydraulic packages, the control electronics and lubrication pumps. A removable aluminum fairing enclosing the nacelle for weather protection has louvers for air cooling and provides mounting for wind sensors.

e. Yaw drive. - Rotation is provided by the yaw drive system, consisting of upper and lower structures, a cross-roller bearing, dual hydraulic motors and hydraulic brakes as shown in figure 10. The yaw brakes control dynamic excitations by maintaining a rigid connection while the nacelle is stationary and also assist in damping yaw motions by maintaining a holding force while the nacelle is being rotated. Power and signal data are transferred to tower mounted cable by slip rings.

f. Tower. - The steel tubular truss tower as shown in figure 11 is made of seven vertical bays. Tubular members were used to reduce "tower shadow" loads on the blades as they pass the tower. Access to the yaw drive and nacelle area is provided by a cable guided, gondola-type lift also shown in figure 11.

g. Ground equipment. - The major ground equipments are the control enclosure, station battery system and the 4.16/12.57 kV stepup transformer. The enclosure, measuring 28 x 10 feet is an air-conditioned steel structure which contains power equipment switchgear and the WTG control and recording unit.

#### PROGRAM STATUS

During the summer of 1978 the WTG without blades was assembled at the GE Riverside facility in Philadelphia. As shown in figure 12, the yaw drive, nacelle structure, drive train, generator and hub with the blade

pitch change mechanism were mounted on the upper section of the tower which served as a test fixture. The control enclosure, control electronics, switchgear, and computer were also assembled. An auxiliary 200 hp motor was mounted on the nacelle structure above the low-speed shaft to rotate the WT drive train and rotor during test. The NASA Portable Instrumentation Van was used to record data from the Engineering Data Acquisition System Sensors.

The factory test program consisted of a checkout of the lube and hydraulic systems and the operation of the yaw drive and pitch change mechanism. The yaw drive rotated 360° and the brake system operation was demonstrated. The pitch change mechanism was operated from the maximum power position to full feather. The rotor was driven at rated rpm, but at a reduced power level, for 20 hours with intermittent yaw maneuvers and pitch change operations. The power generation system was checked out with generated power being dissipated in a load bank.

After test completion in October of 1978 the WTC was disassembled into subassemblies for shipment to the site. Most components were either oversize or overweight for normal road transportation. All subassemblies were shipped by motor vehicle, however, some required special permits. The hub and pitch change mechanism assembly which was shipped by rail due to its weight of 96,000 pounds was the one exception.

Howard's Knob at Boone, NC, is the site selected by DOE for the MOD-1 WTC. The elevation of this site located in northwest NC is 4420 feet above sea level. The Blue Ridge Electric Membership Corporation (BREMC), a rural electric cooperative, will operate and utilize the power generated by the WTC. BREMC is the largest cooperative in North Carolina with annual sales of 555 million kW-hr. BREMC with a peak load of 136 megawatts purchases essentially all of its power for its members from Duke Power.

The Howard's Knob site overlooking the college town of Boone was cleared of trees and graded during the summer of 1978. The concrete tower foundations with the control enclosure, tower lift and transformer pads were poured during August 1978. During October the tower was erected in three sections using a Manitowoc 4100N crane with a boom height of 230 feet and a lift capacity of 55 tons. The WTC installation began in November and consisted of a series of lifts. One lift was considered but was found not to be cost effective and would have had significant schedule risk due to the limited availability of cranes with 200 tone capacity. The installation of the WTC without blades was completed in December and can be seen in figure 13. Figure 14 is a closer view of the WTC with the fairing in place. Shortly after the WTC was assembled aloft, the control enclosure was installed beneath the tower.

Site activity from mid-December to mid-February was curtailed due to extreme cold (wind chill factors of -50° F), high winds (60 mph), icing on the WTC and snow which made the site inaccessible.

Currently the WTG is fully assembled except the blades which are expected to be delivered in April. All cables have been pulled, terminated and checked out. The machine has been mechanically checked out in a similar manner to the Riverside tests, and control/software integration has been in progress since March.

#### CALCULATED OPERATION CHARACTERISTICS

The steady-state operating characteristics are derived from the MOD-1 performance curve,  $C_p$  vs  $\lambda$  (fig. 15). Calculations of the operating characteristics were based on power rating of 2000 kW<sub>e</sub>, a rotor diameter of 61 m (200 ft), and a rotor speed of 35 rpm. The MOD-1 design rpm was determined by maximizing annual energy capture ( $6.5 \times 10^6$  kW-hr) at sea level with 100% availability in an 18 mph (mean) wind regime. Using the  $C_p$  curve, the electrical power output is calculated as a function of wind speed (fig. 16) which establishes the steady state operating requirements for pitch control and the operating wind speeds for generator cut-in and rated power. The breakaway wind speed is based on calculations of the minimum static blade torque required for starting.

The MOD-1 operating envelope (fig. 17) indicates the operational modes and limits for variations in wind speed and direction. The non-operating mode is shown below the cut-in wind speed of 11 mph. A 5-minute average wind speed and yaw angle above 11 mph and 5° respectively will initiate a yaw maneuver, as shown. Normal operation is obtained when the 5-minute average yaw angle is within the 5° envelope. Normal shutdown is initiated when the 5-minute average wind speed exceeds 35 mph or exceeds the wind speed-yaw angle envelope as shown in figure 17. The emergency shutdown mode is initiated when instantaneous wind speeds and yaw angles exceed 40 mph or 90°, respectively.

Calculations of the system dynamic operating characteristics are based on inherently conservative assumptions of statistical wind dynamics and resulting dynamic interactions with the wind turbine. Resulting operating characteristics in terms of critical operational modes, control functions, and electrical stability are shown in table III.

#### COST OF ELECTRICITY/COST DRIVERS

As the first of the megawatt class wind turbines, the MOD-1 was designed to insure long life, reliability and safe operation with current state-of-the-art technology. The resulting cost of electricity is expected to be high on the "learning" curve and reflects the inherent design conservatism indicated by subsystem costs and weights. Therefore, the principle cost drivers are the subsystem weights, a lack of maturity in blade design and fabrication, and a lack of experience in assembly, erection, and testing of the system.

A breakdown of the MOD-1 weights and costs of electrical energy by subsystem are shown in table IV to aid in identifying the significant cost

drivers. The cost of electricity (COE) is derived for each subsystem, based on an annual FCR of 18%, an annual energy capture with 90% availability in an 18 mph (mean) wind regime, and includes the cost-of-doing-business in the cost of each subsystem. An annual operating maintenance cost of 1% is conservatively assumed as reasonable for the early "prototype" systems.

#### DESIGN TRADE-OFFS

The MOD-1 Configuration was primarily dictated by the NASA-LeRC design specifications as previously discussed. Some configuration options were left open for design tradeoffs. The procedures used to evaluate these options were generally tradeoffs between performance, structural design requirements, and cost. A brief description of the tradeoff procedure and results for each option is shown below:

Blade airfoil - Performance vs manufacturability/cost. Airfoil selection driven by manufacturability. Selected 44XX series.

Blade twist - Performance vs blade loading. Blade twist driven by structural design requirements. Selected 11 degrees.

Rotor speed - Maximum energy capture vs torque, cost. Rpm driven by maximum energy capture for a given rotor diameter, rated power, wind duration curve. Selected 35 rpm.

Rotor cone angle - Balance blade thrust - centrifugal loads. Cone angle selected to minimize blade root stress. Selected 9°.

Rotor axis inclination - Blade clearance vs yaw moments. Rotor coning more effective. Selected 0° axis inclination.

Hub (rigid vs teetered) - Blade-hub load reductions vs cost. Rigid hub less costly. Selected rigid hub.

#### FACTORS AFFECTING THE DESIGN

On the MOD-1 Program one of the most significant factors affecting the design was the Technical Specification. During the preliminary design phase a few of the requirements were modified to reduce the WTG costs. The double bearing shaft of the drive shaft/rotor support was replaced by a single bearing with reduced weight and cost. For the yaw drive, a hydraulic motor was used instead of the electric motor-driven worm gear which resulted in less space and weight and better overload protection. Reduced cost was also obtained for the blade inching drive by replacing the independent drive on the high speed shaft with a blade operational control system.



Prior to the establishment of the final design, a rigorous requirements assessment analysis was conducted in an effort to minimize requirements and, hence, reduce WTG costs. At this late stage of the design process the opportunities were limited; however, critical design parameters were modified to reduce WTG costs and the cost of generated electricity. For example, epoxy/glass was replaced with steel as the blade structural material. The rated power was increased from 1500 to 2000 kW as a system limit (present blades have a limit of 1818 kW) in order to increase the rating and energy capture. The cut-out speed was reduced from 50 to 35 mph. This decreased the blade and system loads with only a minor loss (45%) in energy capture. Furthermore, the blade tip clearance was reduced from 50 to 35 feet in order to lower the tower height. This reduced cost and system loads with a minor (3%) loss in energy capture.

The load requirements were also modified to realistically include the effects of accumulative fatigue over the entire wind regime. In addition, the load cases were simplified to four cases which included continuous, gust, emergency feather and hurricane loads, as shown in table V. To make the gust loading more realistic the wind gust model was modified per figure 18 which replaced the earlier 1-cosine curve.

After the requirements and the design concept were solidified, the sizing and detail of each component were dictated by certain design parameters. The most significant design drivers for each major component are shown in table VI. As one can observe, fatigue and stiffness have driven the weight of the mechanical configuration. Stiffness has played a prominent role in sizing the pitch change mechanism and the 3.2 P tower, and consequently has affected costs. Limit loads have played a secondary role in dictating system weights.

#### DESIGN EVALUATION

As mentioned earlier, the MOD-1 WTG is the first of the megawatt size WTG's. With the reservations that we do not have any operating experience at this time, some overall conclusions about the MOD-1 WTG design can be made:

- The design is conservative.
- The weight and cost are high.
- The installation is routine.
- The extensive instrumentation should provide design data for future WTG's.

#### RECOMMENDATIONS FOR FUTURE DESIGNS

Our current recommendation for a future design is the result of a NASA-directed MOD-1A trade-off study. The objective of the study was to reduce

weight and cost of a 2 megawatt WTG with the same operational characteristics as the MOD-1 without restrictions on the design concept. The objectives of the study were to: reduce weight from 655,000 to 400,000 pounds or less; reduce second unit cost from \$2,900/kW to \$1,000/kW; and reduce the cost of energy from 18¢/kW-hr to 5¢/kW-hr (all costs in 1978 dollars). The design approach was to "wage war on weight" by loads alleviation and simplification. Three candidate concepts, shown in figure 19, were considered for trade-off studies of critical design parameters.

System Number 3 of figure 19 was selected which has as its major characteristics a teetered hub, two downwind blades with partial span control, an integral gearbox structure, an inclined rotor axis and a "soft" tower. Figure 20 is an outline drawing of the MOD-1A. The selected blade has a MOD-1 aerodynamic configuration except that the concept of hydraulic driven partial span torque control is incorporated in the outer 15% of the span. The teetered hub concept resulted in the lowest loads for a two-blade system. The gearbox/bedplate incorporates the rotor and yaw support structure into the gearbox casing, thus eliminating structural weight. The tower is a conical shell with a lateral bending frequency of 1.2 P.

An overall comparison of the MOD-1 and 1A can be seen from the silhouette of -1 superimposed over the MOD-1A in figure 20. This comparison illustrates the striking reduction in size of the MOD-1A. The most impressive statistic is the magnitude of the weight reduction shown in figure 21. WTG costs, as a consequence, are reduced dramatically, and it follows that the cost of generated electricity is reduced accordingly. The projected installed cost in 1978 dollars of the MOD-1A is in the neighborhood of \$1050/kW. As a result, the cost of energy has been reduced to 6¢/kW-hr which is a significant improvement when compared to earlier WTG's, as shown in figure 22. In summary, the MOD-1 will serve the purpose of supplying valuable WTG operating data for the national wind program and the concepts of the MOD-1A will lead us to commercially viable WTG's.

#### DISCUSSION

- Q. Have you investigated designing a machine with a soft tower? What technical risks, if any, are associated with a soft design?
- A. This was considered in the slides on our recommendations for the future that were not presented. A conceptual design study, directed by NASA, was conducted in 1977 after the MOD-1 design was finished. In essence, we evolved some concepts that we thought could reduce cost. The soft tower was one of them. We also considered the concept of using an integral gearbox, where the gearbox provides the basic structural member on the tower. We also recommended partial span control to reduce pitch change mechanism costs.



- Q. Your normal operation was shown as a  $\pm 5^\circ$ . Do you feel this is a very close angle?
- A. That value was a 5-minute average, not an instantaneous value. I think the variation was up to  $15^\circ$ . That is the way the system is now programmed to operate, and we will find out from actual experience if that is the effective way to operate the system. Based upon all of the loads that we can measure and the flexibility of using a computer-based control system, we can then make changes in the software and alter that operation.

TABLE I. SUMMARY OF TECHNICAL SPECIFICATIONS

<u>ITEM</u>	<u>REQUIREMENT</u>
RATED POWER	1500 kW <sub>e</sub> @ 22 MPH
CUT-IN WIND SPEED	11 MPH
CUT-OUT WIND SPEED	35 MPH
MAXIMUM DESIGN WIND SPEED	150 MPH (AT ROTOR CENTER LINE - NO WIND SHEAR)
ROTORS/TOWER	1
LOCATION OF ROTOR	DOWNWIND
DIRECTION OF ROTATION	CC (LOOKING UPWIND)
BLADES PER ROTOR	2
CONE ANGLE	OPTIONAL
INCLINATION OF AXIS ROTATION	$\leq 15^\circ$
ROTOR SPEED CONTROL	VARIABLE BLADE PITCH
ROTOR SPEED	OPTIONAL/CONSTANT
BLADE DIAMETER	200 FT. (NOMINAL)
AIRFOIL	OPTIONAL
BLADE TWIST	OPTIONAL
TOWER	STEEL TRUSS
BLADE TIP TO GROUND CLEARANCE	> 50 FT.
HUB (RIGID VS. TEETERED)	OPTIONAL
TRANSMISSION	FIXED RATIO GEAR, 96% EFFICIENCY
GENERATOR	60 Hz/SYNCHRONOUS
YAW RATE	$\leq 2^\circ/\text{SEC}$
CONTROL SYSTEM	ELECTRO MECHANICAL/ MICROPROCESSOR

TABLE II. BLADE DESIGN LOADS

<u>CASE NUMBER</u>	<u>DESCRIPTION</u>	<u>FREQUENCY OF OCCURRENCE</u>
1	RATED POWER, RATED WIND SPEED	$10^8$
2	INITIALLY AT RATED POWER, WIND SPEED INCREASE FROM RATED TO 60 MPH IN 1/4 SEC, NO PITCH CHANGE, ROTOR OVERSPEED 25%.	$10^5$
3	INITIALLY AT RATED POWER, CHANGE PITCH ANGLE TO FEATHER IN 11 SECONDS.	OCCASIONAL (PROPORTIONAL LIMIT)
4	INITIALLY AT RATED POWER, WIND SPEED DECREASED FROM RATED TO ZERO IN 1/4 SECOND.	$10^5$
5	BLADES IN HORIZONTAL FEATHERED POSITION: WIND SPEED 120 MPH FROM ANY DIRECTION.	OCCASIONAL (PROPORTIONAL LIMIT)
6	ROTOR OPERATING AT DESIGN RPM, WIND SPEED 50 MPH AT 20° YAW ANGLE, CHANGE YAW ANGLE @ 20°/SEC.	$10^5$
7	ROTOR OPERATING AT DESIGN RPM, NO POWER, VELOCITY RETARDATION OF 50% DUE TO "TOWER SHADOW"	$10^5$

TABLE III. - DYNAMIC OPERATING CHARACTERISTICS

<u>Item</u>	<u>Characteristic</u>
<u>CRITICAL OPERATING MODES:</u>	
1. WIND VARIABILITY:	<ul style="list-style-type: none"> <li>• GUSTING - MAGNITUDE/DURATION (RANDOM)</li> <li>• DIRECTIONAL (RANDOM INFLOW)</li> </ul>
2. CYCLIC BLADE LOADING:	<ul style="list-style-type: none"> <li>• TOWER</li> <li>• WIND SHEAR</li> <li>• WIND INFLOW</li> </ul>
3. NON-OPERATING:	<ul style="list-style-type: none"> <li>• CUT-OUT</li> <li>• LOSS OF LOAD</li> <li>• FEATHERING</li> </ul>
<u>CONTROLS AND RESPONSE</u>	
1. PITCH CONTROL:	<ul style="list-style-type: none"> <li>• 2.1 °/SEC OPERATING/0.2 SEC RESPONSE</li> <li>• 14 °/SEC (MAX. EMERGENCY FEATHER)</li> </ul>
2. YAW CONTROL:	<ul style="list-style-type: none"> <li>• 15 °/MIN.</li> </ul>
3. SLIP CLUTCH:	<ul style="list-style-type: none"> <li>• @ 15,400 FT-LBS (188% RATED TORQUE)</li> </ul>
<u>ELECTRICAL STABILITY</u>	
1. CALCULATED TORQUE/SPEED VARIATIONS:	<ul style="list-style-type: none"> <li>• 420,000 FT-LBS (+ 10%)</li> <li>• 35 RPM (+ 2%)</li> </ul>
2. CALCULATED ELEC. POWER VARIATIONS:	<ul style="list-style-type: none"> <li>• ± 6% (CYCLIC)</li> <li>• ± 30% (MODERATE GUSTS)</li> <li>• ± 100% (MAX. GUSTS)</li> </ul>
3. CALCULATED VOLTAGE VARIATIONS:	<ul style="list-style-type: none"> <li>• MODERATE GUSTS (&lt;5% @ GEN. TERMINALS)</li> <li>• MAXIMUM GUSTS (&lt;10% @ GEN. TERMINALS)</li> </ul>

TABLE IV. - COST OF ELECTRICITY  
(MOD-1 2ND UNIT RECURRING COSTS)

SUBSYSTEM	WT, LBS	COE, ¢/kW-HR	% TOTAL COE
BLADES	41,000	4.6	25%
HUB	44,000	1.3	7
TORQUE CONTROL	23,000	0.6	3
NACELLE/STRUCT. & DRIVE TRAIN	153,000	2.4	13
POWER GEN. EQUIP.	17,000	1.1	6
CONTROLS	1,000	0.7	4
YAW DRIVE	56,000	1.0	5
TOWER	320,000	1.3	7
ASSEMBLY/TEST	-	2.3	12
SITE PREP/ERECT. & CHECKOUT	-	2.4	13
TOTALS	655,000	17.8	95%
ANNUAL O&M	-	0.8	5%
TOTAL COE		18.6	100%

TABLE V. - MODIFIED BLADE DESIGN LOADS

CASE	REQUIREMENT	FREQUENCY OF OCCURRENCE
A	ACCUMULATIVE FATIGUE ENTIRE WIND REGIME 20° INFLOW ANGLE INCLUDED	$4 \times 10^8$ CYCLES
B	35 - 50 MPH GUST 35 - 20 MPH GUST BLADE DISC FULLY IMMersed MODIFIED WIND GUST MODEL NO PITCH CHANGE	$10^5$ CYCLES
C	EMERGENCY FEATHER  <div style="display: flex; justify-content: space-around;"> <div> <u>RPM</u>  <math>N_0 &lt; n &lt; 1.4 N_0</math>  <math>n &lt; N_0</math> </div> <div> <u>PITCH RATE</u>  <math>11^\circ \text{ SEC}</math>  <math>3^\circ \text{ SEC}</math> </div> </div>	$10^5$ CYCLES
D	HURRICANE BLADE FEATHERED IN HORIZONTAL POSITION 120 MPH FROM ANY DIRECTION	OCCASIONAL (PROPORTIONAL LIMIT)

TABLE VI. - DESIGN DRIVERS

SUBSYSTEM	DESIGN DRIVER
BLADES	CUMULATIVE FATIGUE, EMERGENCY FEATHER LOADS AND BLADE STIFFNESS.
R/P	FATIGUE; BLADE WEIGHT AND TORQUE CONTROL MOMENTS.
TORQUE CONTROL	GUST LOADS, MAX. FORCE EMERGENCY SHUTDOWN AND STIFFNESS.
BEARING & DRIVE TRAIN	MAX. AND CYCLIC TORQUE ROTOR LOADS ON BEARING, POWER LEVEL.
NAVELLE STRUCTURE	CUMULATIVE FATIGUE IN WELDS.
POWER GENERATION EQUIPMENT	POWER LEVEL, POWER QUALITY WTG UTILITY PROTECTION.
CONTROLS	UNATTENDED OPERATIONS, POWER QUALITY.
YAW DRIVE SYSTEM	TORQUE (MAX. WINDSPEED & IN-FLW ANGLE) OVERTHAUL MOMENT.
TOWER	LATERAL STIFFNESS AND FATIGUE.
ASSEMBLY & TEST	NO. OF PARTS, JOINTS AND CONNECTIONS CRITICAL ALIGNMENTS AND WEIGHTS.
SITE PREPARATION, ERECTION AND ROCKET	SITE CHARACTERISTICS, LOCATION, WTG WEIGHT AND NO. OF SUBASSEMBLIES.

• DESIGN, FABRICATION, INSTALLATION AND TEST OF A 1500 KW WIND POWER GENERATOR

- STATE-OF-THE-ART TECHNOLOGY FOR MINIMUM TECHNICAL RISK.
- COMPATIBLE WITH LARGE AND SMALL UTILITIES.
- CAPABLE OF UNATTENDED OPERATION -- I.E. AUTOMATIC AND REMOTE CONTROL FROM UTILITY DISPATCH CENTER.
- CAPABLE OF 10 YEAR LIFE, WITH "ROUTINE" MAINTENANCE.
- MINIMUM AVAILABILITY OF 90%.
- SAFE RELIABLE OPERATION.
- SAFE AND EASY MAINTENANCE.
- MINIMUM FIELD ASSEMBLY.
- TRANSPORTATION VIA EXISTING SURFACE VEHICLES.
- SNOW, RAIN, LIGHTNING, HAIL, ICE, SALT VAPOUR, WIND TO 120MPH.
- ACCEPTABLE APPEARANCE.
- COSTS COMPETITIVE WITH ALTERNATE ENERGY SOURCES.

• DEVELOPMENT OF A WTG DESIGN WHICH CAN BE ITERATED INTO A SECOND-GENERATION VERSION SUITABLE FOR HIGH-VOLUME, LOW-COST PRODUCTION.

• ACQUISITION OF DATA AND OPERATION EXPERIENCE WHICH WILL LEAD TO MORE EFFICIENT, SECOND-GENERATION MACHINES.

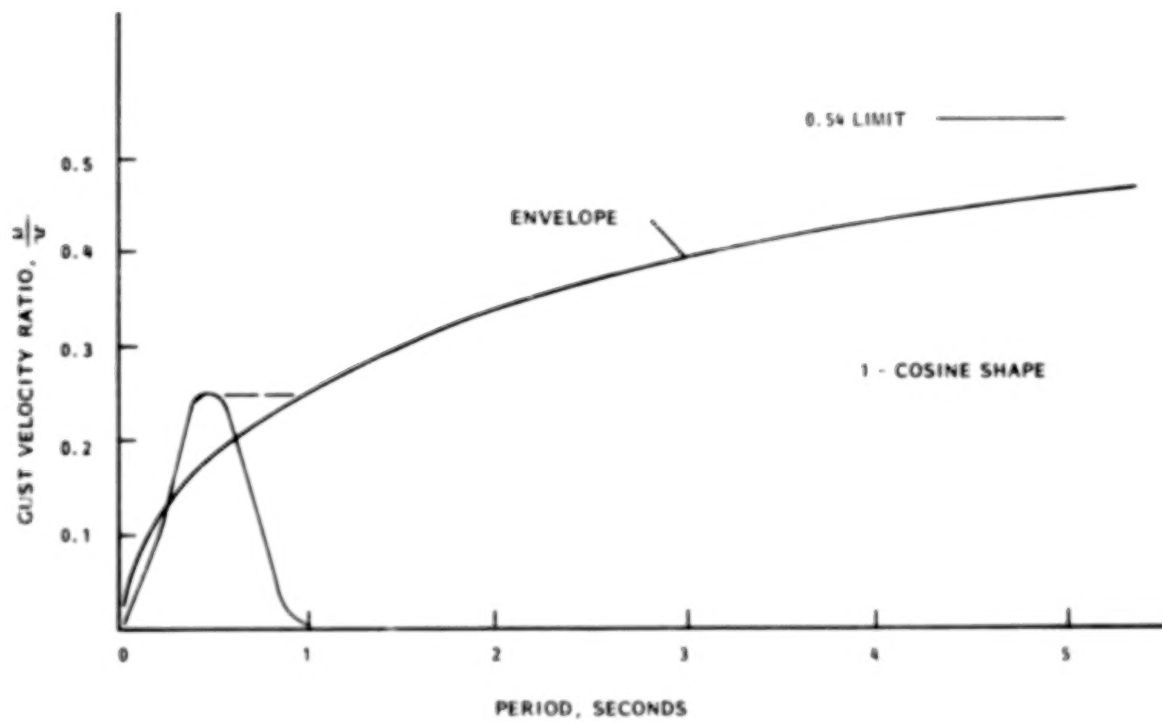


Figure 2. - 1-Cosine wind gust model.



Figure 3. - Scaled model of MOD-1 WTG.

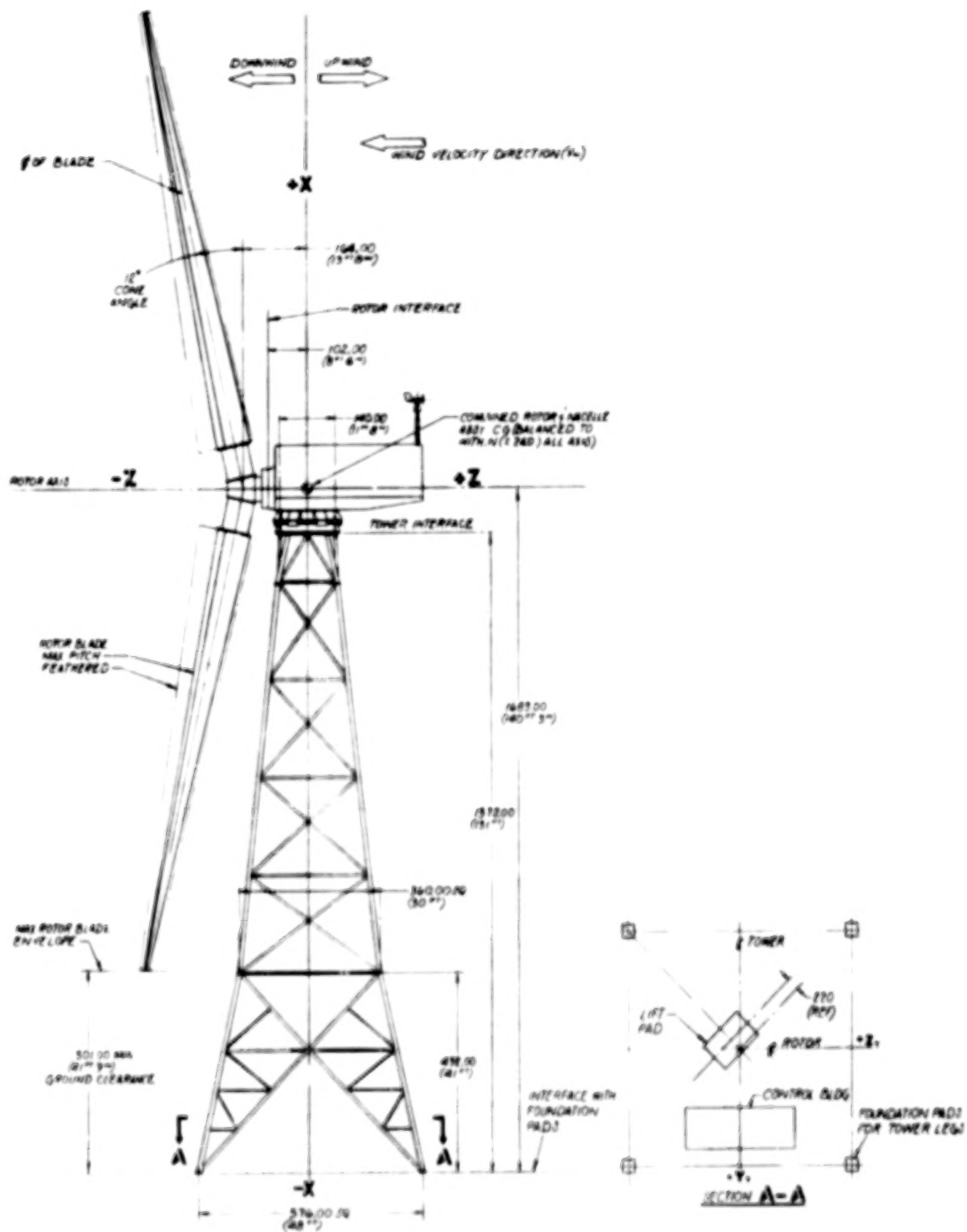


Figure 4. - MOD-1 WTC dimensions.

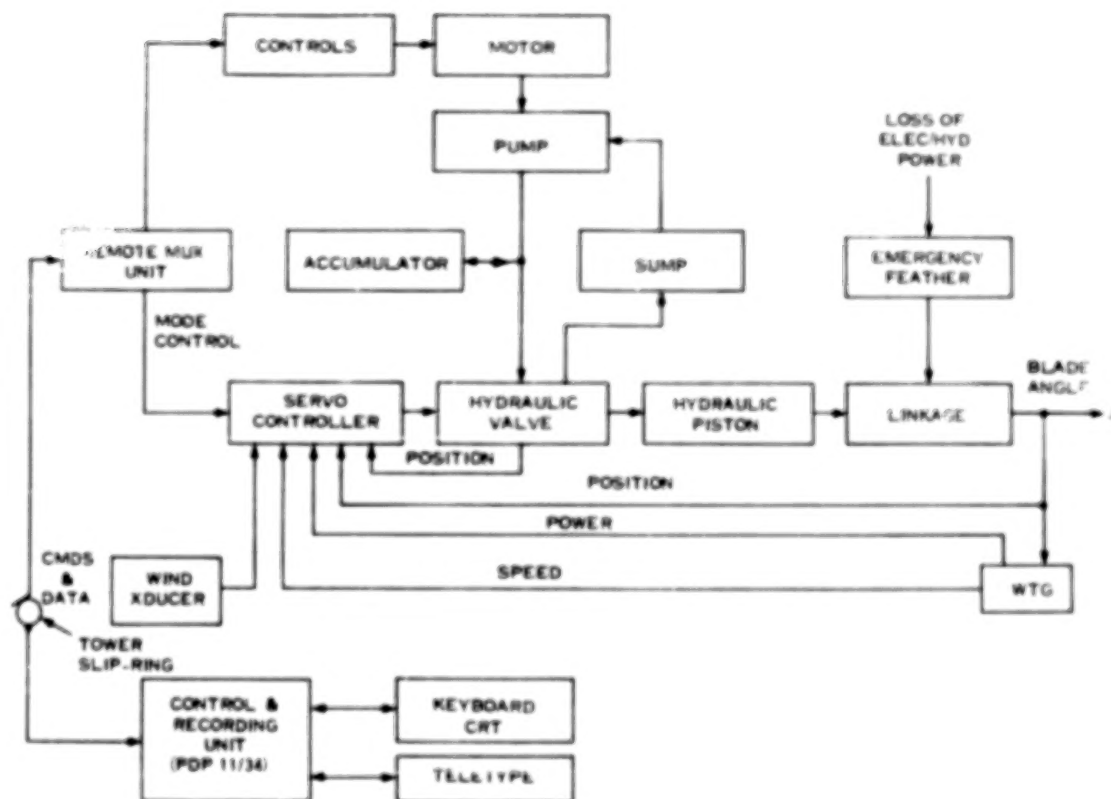


Figure 5. - Blade pitch control diagram.

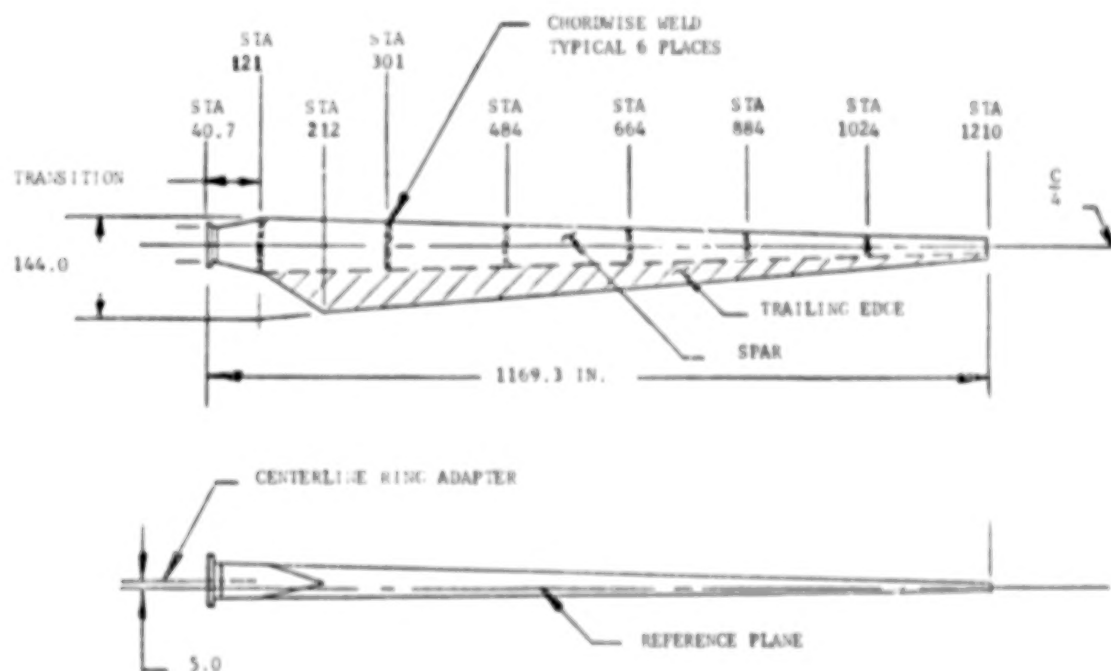


Figure 6. - Blade geometry.





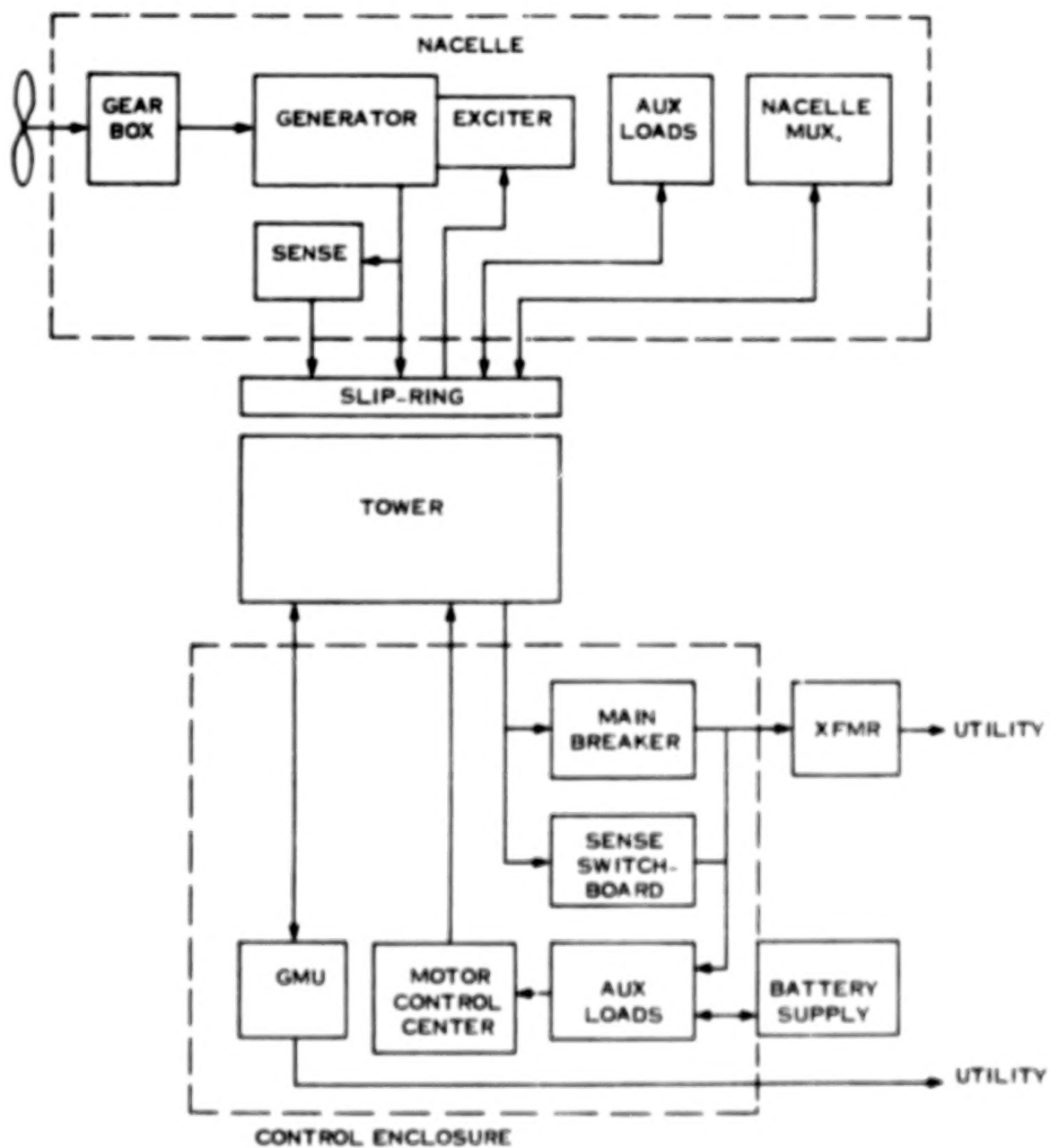


Figure 9. - Power generator/control block diagram.

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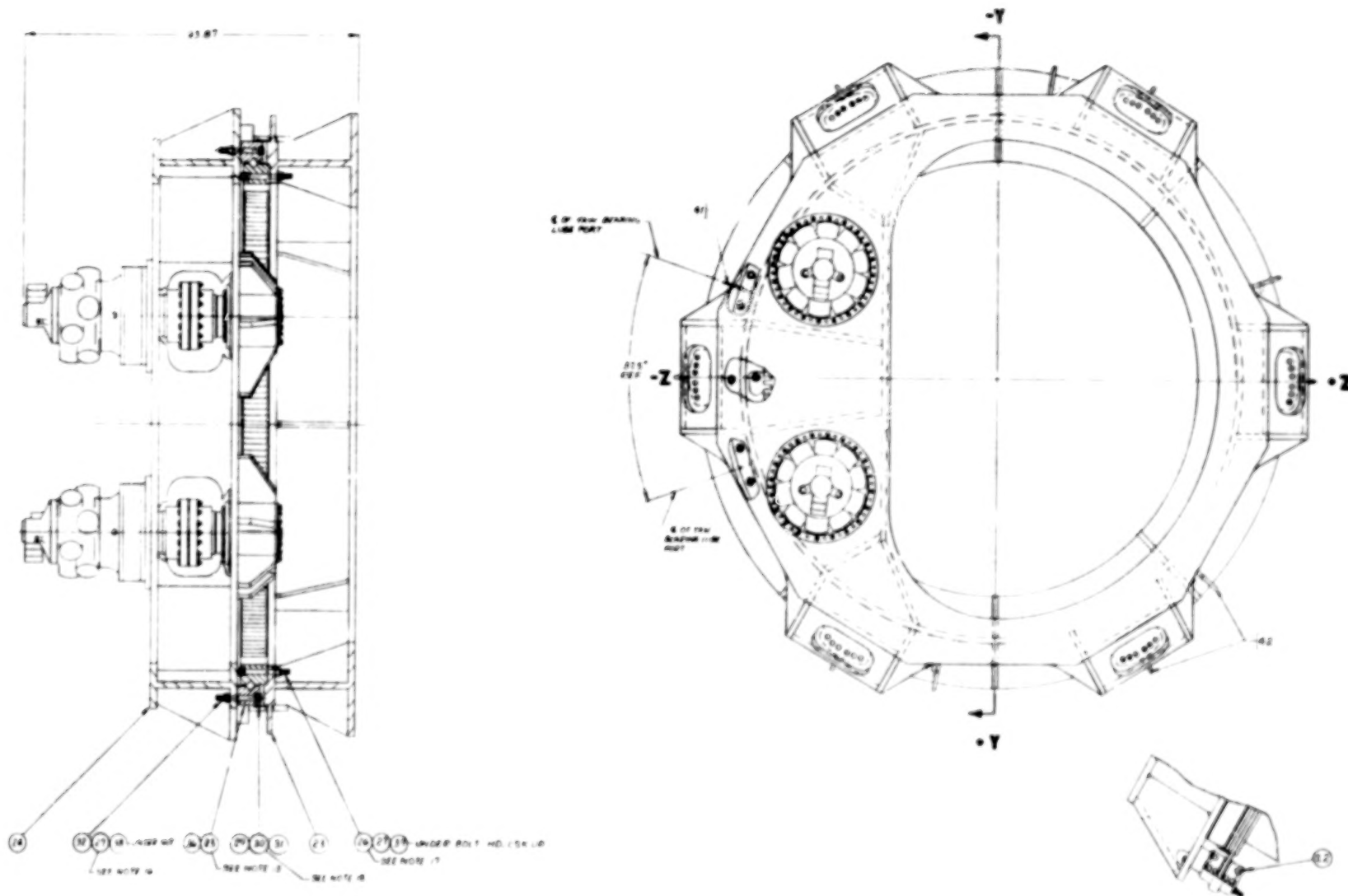


Figure 10. - Yaw drive installation.

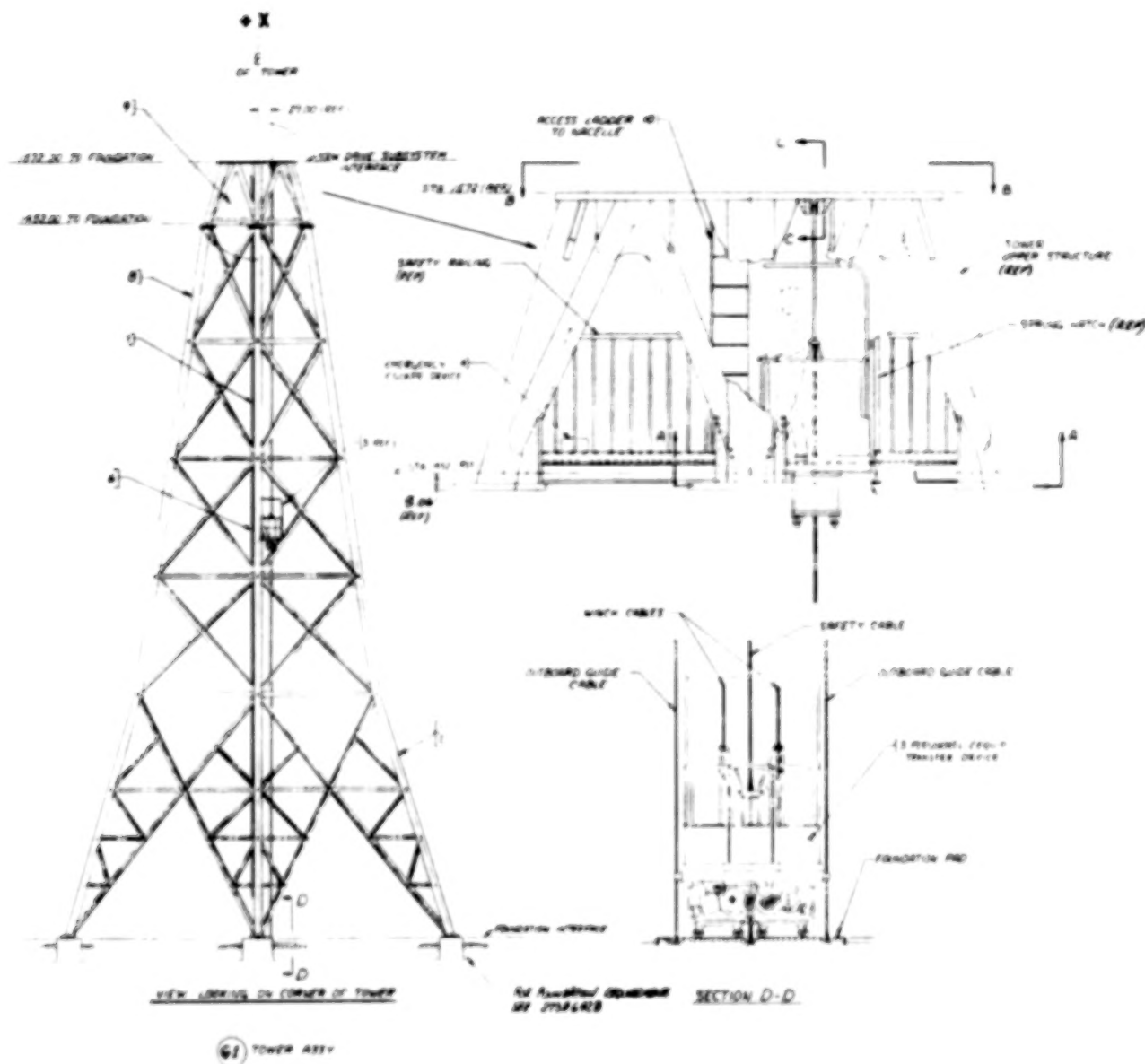


Figure 11. - Tower and lift.

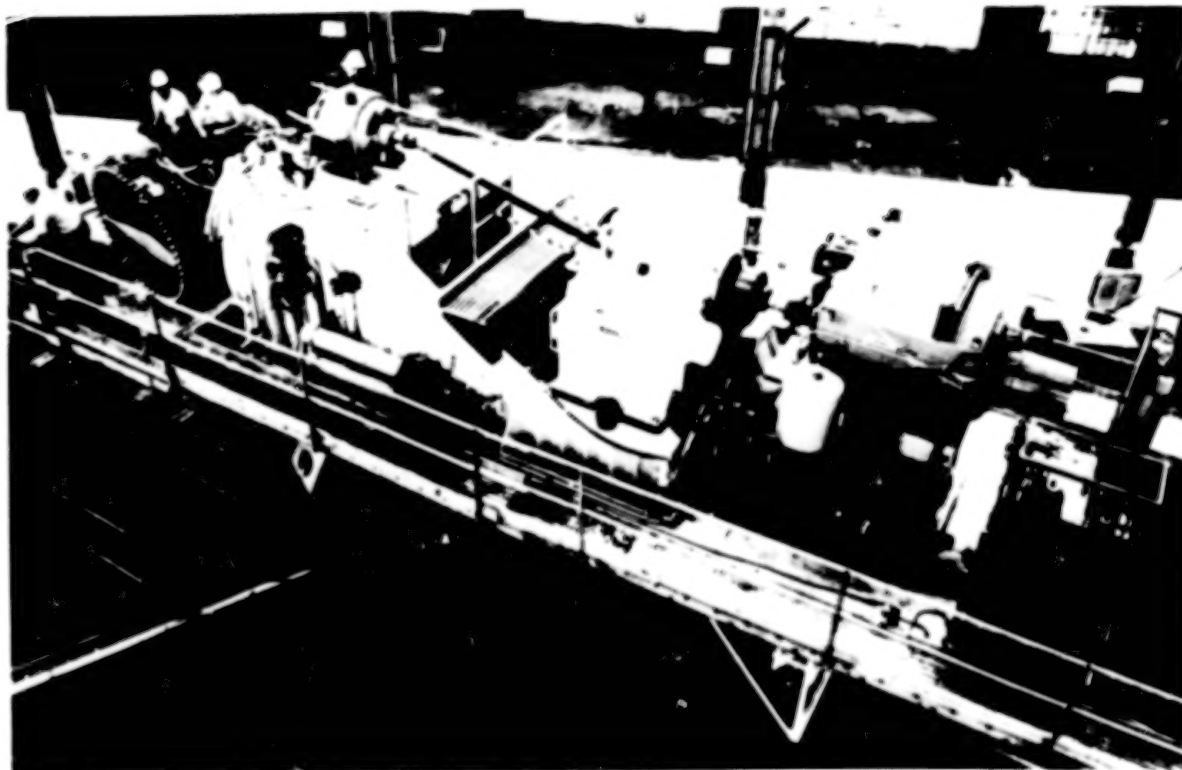


Figure 12. - Test of WTG without blades at Riverside.



Figure 13. - WTG assembly and crane.



Figure 14. - WTG assembled.

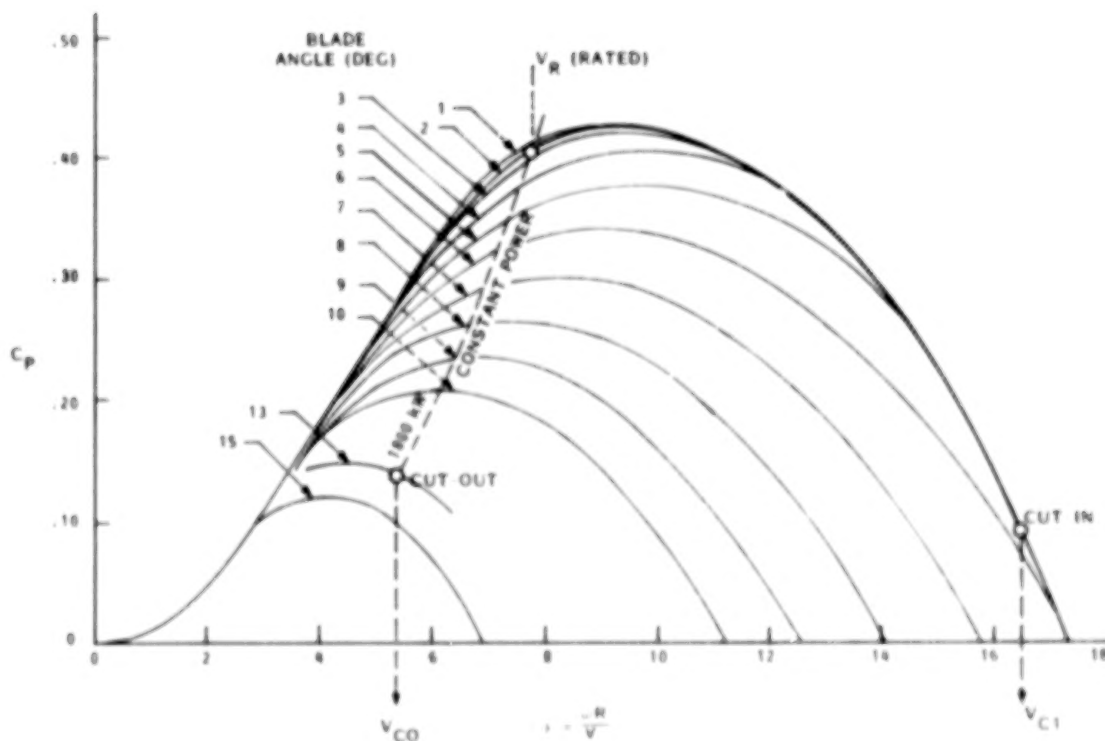


Figure 15. - M10-1 performance curves.

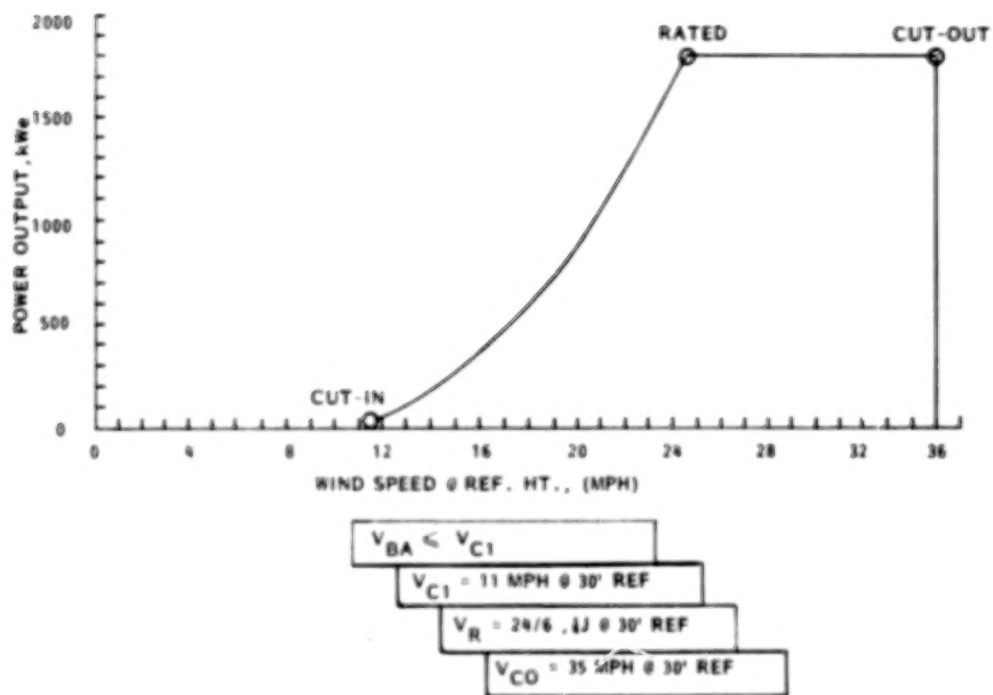


Figure 16. - Steady-state operating characteristics.

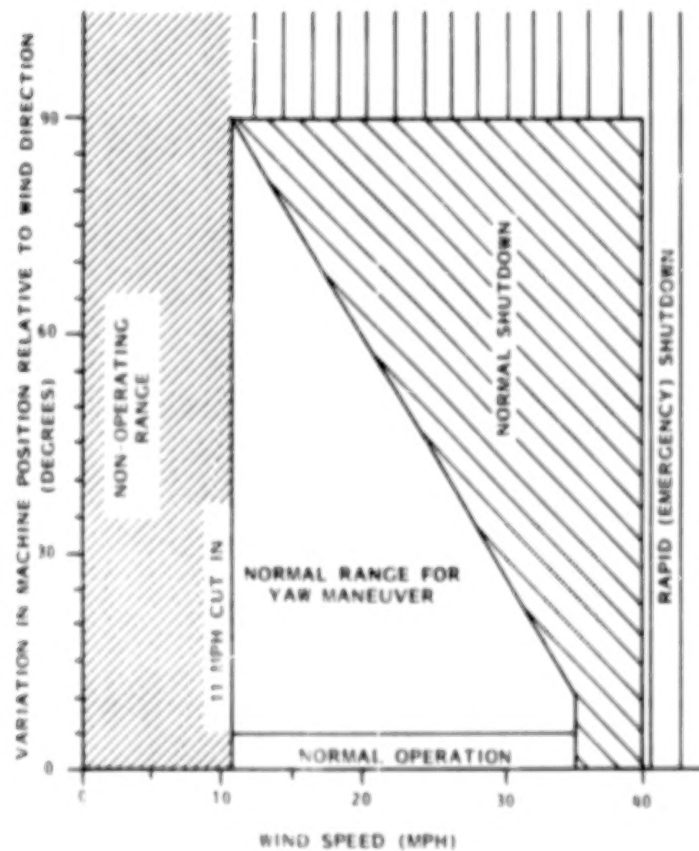


Figure 17. - MOD-1 operating envelope.

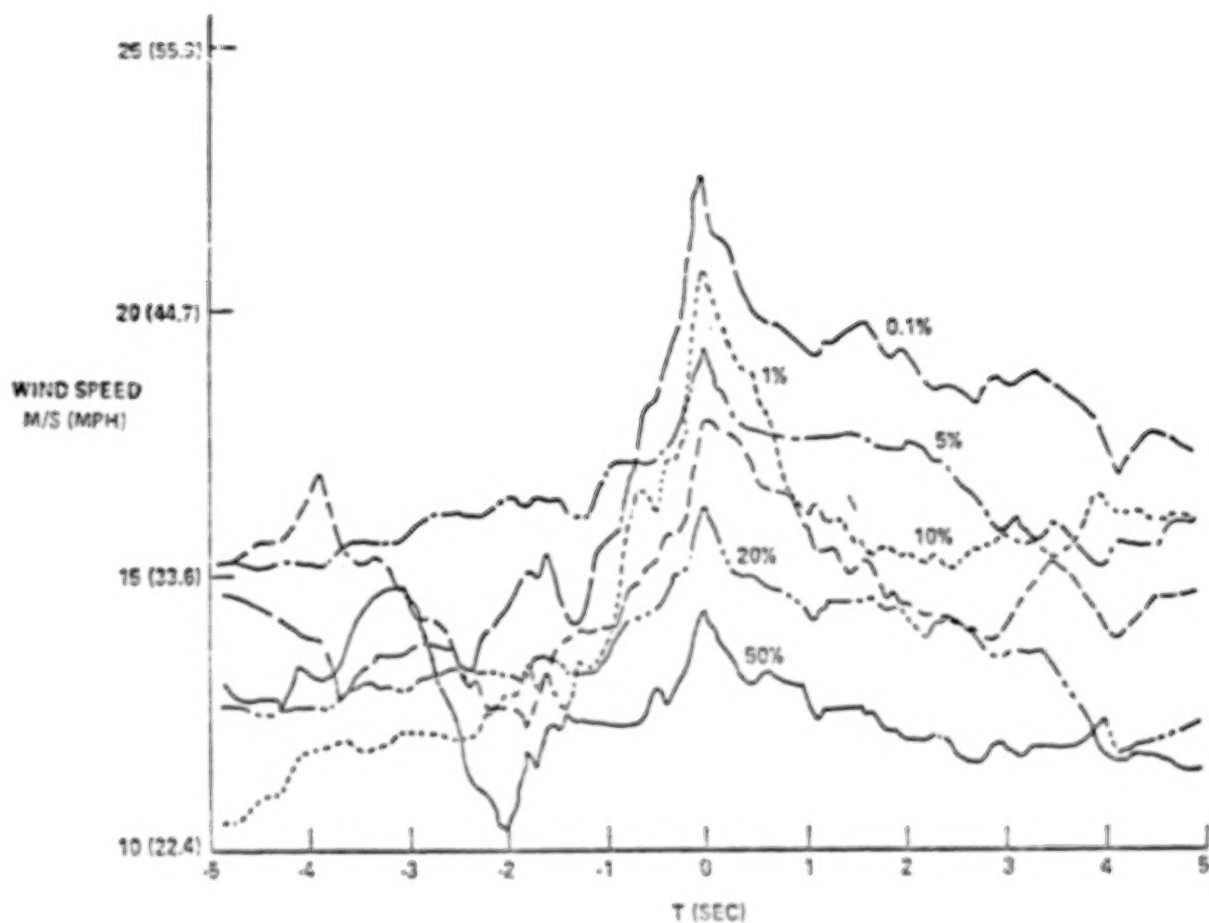


Figure 18. - Modified wind gust model with probability of occurrence.

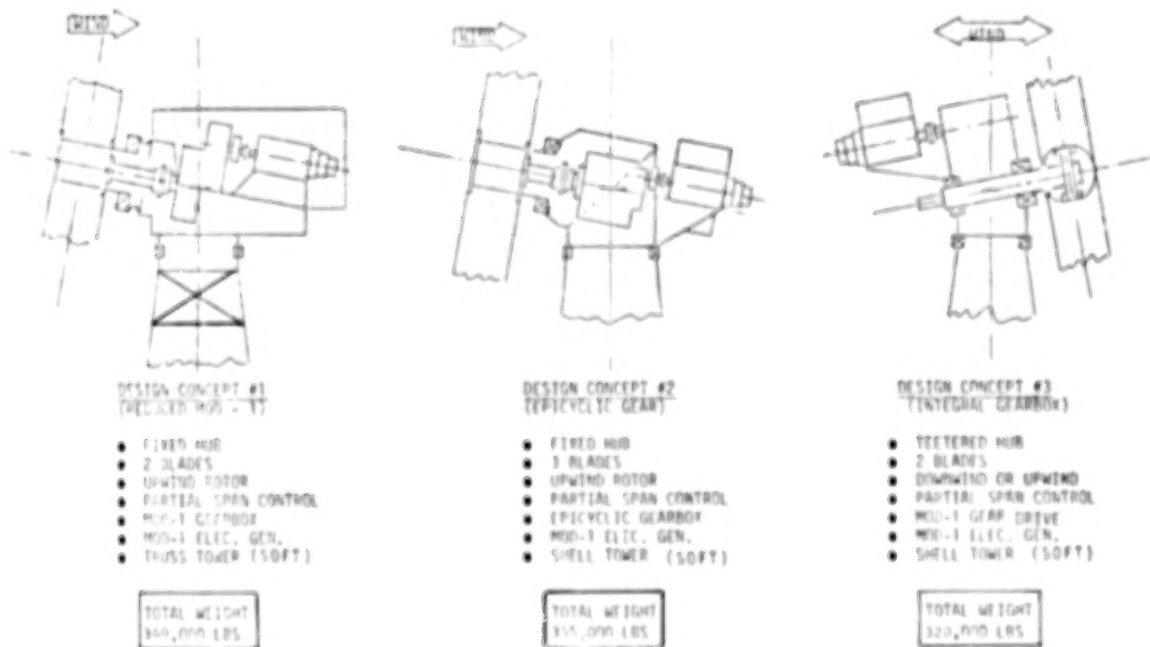


Figure 19. - Three candidate systems.



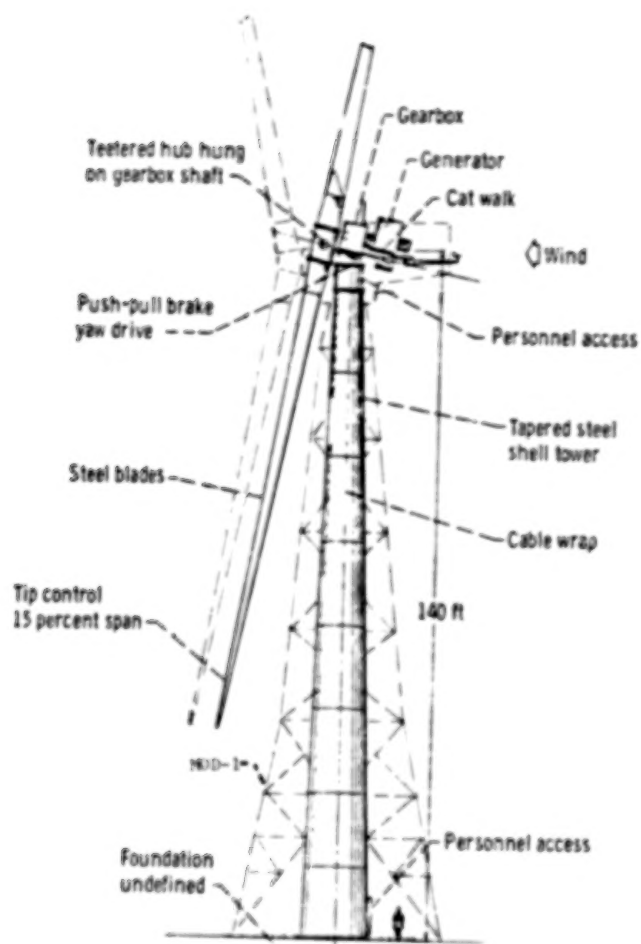


Figure 20. - MDD-1A outline and comparison with MDD-1.

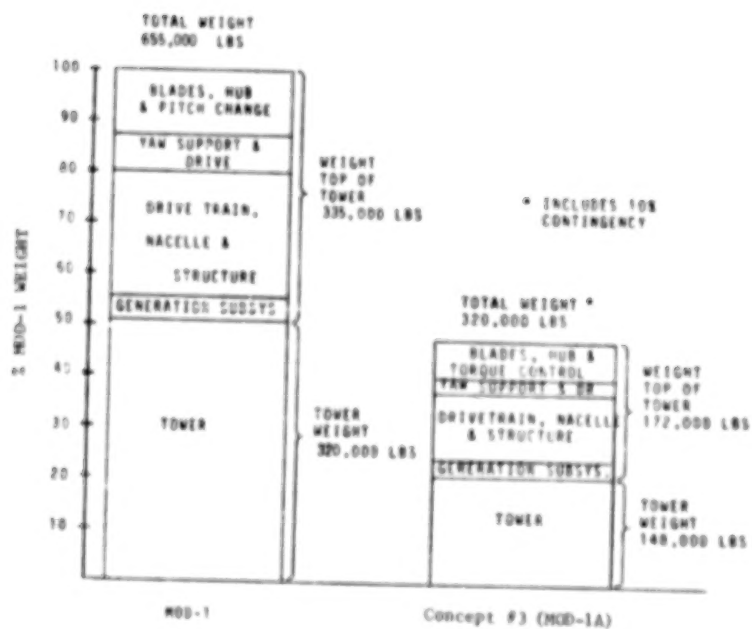


Figure 21. - Weight comparison for MDD-1 and concept #3.

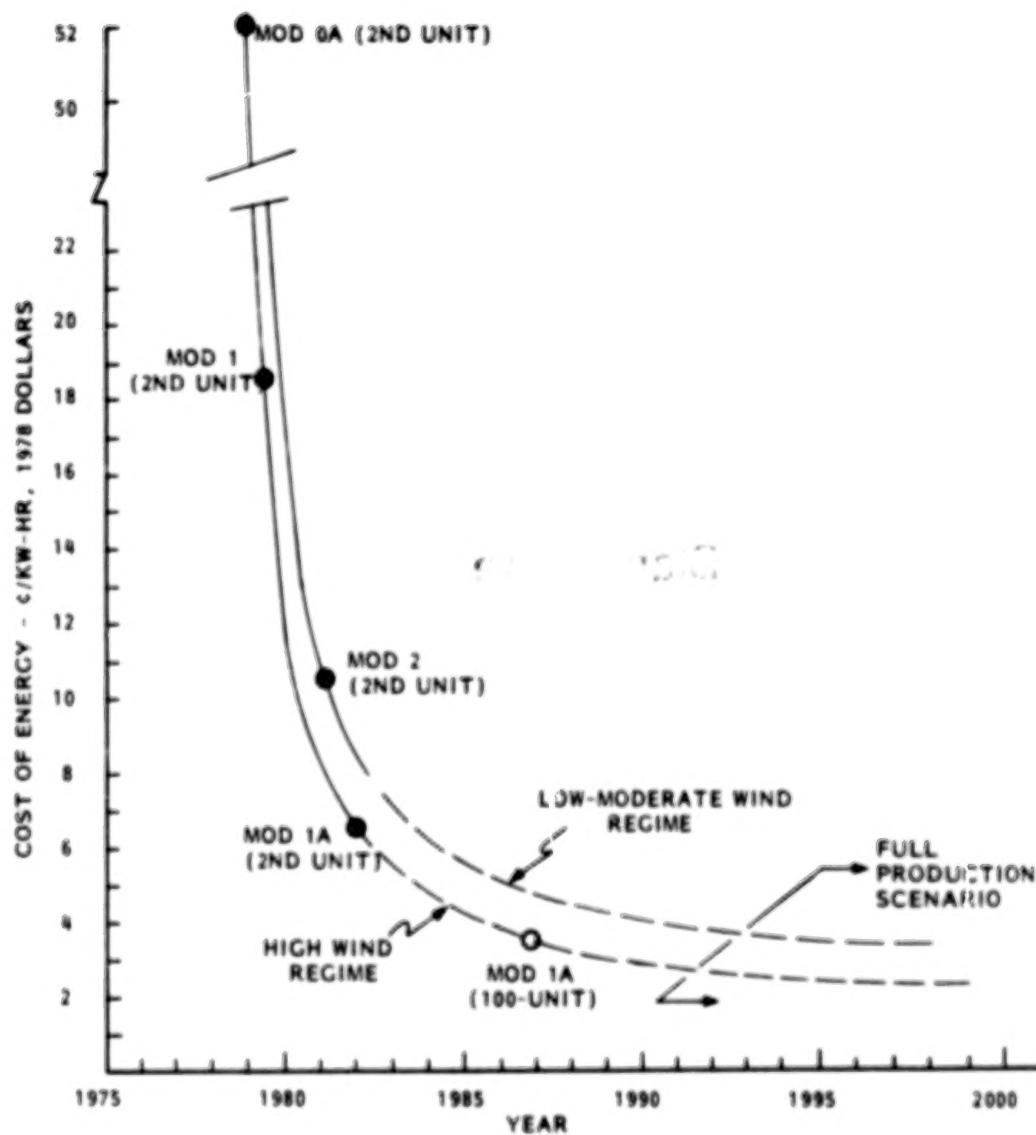


Figure 22. - Cost of energy projections.

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THE BOEING MOD-2 -  
WIND TUNNEL SYSTEM RATED AT 2.5 MW

Richard R. Douglas  
Boeing Engineering and Construction Company  
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INTRODUCTION

The MOD-2 project is an approximate 36 month program for the development, design, fabrication, installation, and check-out of a wind turbine system (WTS) optimized for commercial production of power into a utility grid. Similar to the MOD-0 and MOD-1 programs, MOD-2 is managed by NASA-LeRC. Contrary to those programs, the primary objective of the end hardware is for direct and efficient commercial application, rather than for Research and Development. The program has been structured to achieve this desired commercial objective by a substantial concept selection effort, comparatively few firm requirements imposed on the contractor, and encouragement of commercial practice application. This paper provides a summary description of MOD-2 development and of the resulting system hardware.

PRIMARY SPECIFICATIONS & REQUIREMENTS

The major firm requirements imposed on the contractor were as follows:

- o 14 mph average wind speed at 30 foot altitude.
- o Horizontal axis.
- o Minimum rotor diameter of 300 feet.
- o 30 year service life.
- o Unattended remote site operation.

Essentially all other requirements were subsequently agreed to by NASA and the contractor as a result of requirement sensitivity studies generated during the program concept study phase. A list of the major requirements thus developed are shown in Table I.

PRIMARY DESIGN CHARACTERISTICS

Four significant changes from the MOD-0 and MOD-1 wind turbine system design characteristics were incorporated into the original MOD-2 proposal:

- o Use of a soft shell type tower.
- o An epicyclic gear box.
- o A quill shaft to attenuate 2/rev. torque and power oscillations.
- o A rotor designed primarily to commercial steel fabrication standards.

Through the many months of detailed study since the proposal, these four features are still retained and account for a major portion of any cost-of-electricity advantage that MOD-2 may have compared to competitive systems. During the concept study phase, decisions were made to change from a combination welded and bonded rotor to an all-steel rotor, to use a teetered in place of a fixed hub rotor, to use tip control rather than full span control, to orient the rotor upwind rather than downwind, and to change from a ground located computer with nacelle located multiplexer to a microprocessor system located in the nacelle. Each of these changes resulted in a favorable decrease in cost-of-electricity.

The major characteristics and general arrangement of the current MOD-2 WTS configuration are shown in Figure 1. Illustrations of all other major components of the system are provided by Figures 2 through 12. Weight status is shown in Figure 13. MOD-2 is a horizontal axis machine with a 300 foot diameter, tip control, teetered, upwind rotor. The rotor axis is located 240 feet above ground level. The all steel rotor is supported by the low speed shaft through an elastomeric bearing that permits teetering. Torsion from the rotor is transmitted by an attenuating quill shaft to the step up planetary gear box, which in turn drives a 2500 KW synchronous generator at 1800 rpm. Teeter and rotor brakes are used primarily to eliminate motion when the wind turbine is not operating. All of the drive train, the generator, the generator accessory unit, the electronic control system, the pitch and yaw hydraulic system, and other support equipment are housed in the nacelle. The nacelle itself is kept oriented into the wind by a single hydraulic motor driving through a planetary reduction gear. The tower is a shell type with a conical base and contains an elevator, an emergency ladder, and control and electrical system components in the base. The foundation is conventional reinforced concrete but has a unique inverted mushroom configuration that permits use of earth fill to reduce the concrete required.

#### OPERATING CHARACTERISTICS

The MOD-2 system is designed to operate unattended into a utility grid whose power substantially exceeds the 2.5 megawatt output of MOD-2. The system is designed to cut-in at a wind speed of 14 mph, to cut-out at 45 mph, and to generate full rated power (2.5 megawatts) at 27.5 mph. (See Figure 14) While the MOD-2 system was optimized for a site with an annual mean wind speed of 14 mph at 30 foot altitude, Figure 15 illustrates that it operates with little penalty at sites with a wide band of wind speeds.

Rotor and system efficiencies are best portrayed by the rotor and system coefficient versus wind speed chart shown in Figure 16. The rotor and power coefficients are that portion of the wind's kinetic energy passing through the rotor disk that is converted into torque and electrical energy, respectively. The difference between the two represents the losses in the turbine subsystems.

#### COST OF ELECTRICITY

Cost-of-electricity assessment for MOD-2 is based on cost of the 100th production unit. Fig. 17 illustrates the cost approach and Fig. 18 presents the cost groundrules and resulting costs. The cost-of-electricity is computed as follows:

$$\text{COE} = \frac{\text{IC} \times \text{FCR} + \text{AOM}}{\text{AEP}}$$

where IC = total WTS cost = \$1,720,000

FCR = annualized fixed charge rate = 18%

AOM = annual operation and maintenance = \$15,000

AEP = annual energy production =  $9.75 \times 10^6$  kWh

COE = 3.3 ¢/kWh

#### PROGRAM APPROACH

During the Third Wind Energy Workshop in Sept., 1977, Jim Couch did a fine job of describing the MOD-2 planned design approach. Briefly, this consisted of a substantial conceptual design effort to select the most cost effective system concepts, a preliminary design effort to refine the design and a detail design phase to produce the final drawings. At this time, we are nearing completion of the detail design phase. In fact, numerous releases have already been made for long lead items such as the gear box, low speed shaft bearings, yaw bearings, etc. With the exception of some contract extensions during the concept and preliminary design phases to conduct additional studies desired by NASA-LeRC, the program has proceeded as planned. A summary of program events and future plans is illustrated on the schedule shown in Fig. 19.

## MAJOR DESIGN FACTORS

Undoubtedly, the most important of all the design features on MOD-2 is the soft shell type tower concept. Fig. 20 illustrates a comparison between a soft shell type tower and a stiff truss type tower at the time of the original study. The basic difference between the soft tower and stiff tower is shown in Fig. 21, illustrating that the soft tower has a lower frequency than the rotor while a stiff tower has a higher frequency. Fig. 22 illustrates the precise relationship of the tower design. Note that it is designed by a combination of frequency, seismic, fatigue, and high wind factors. Not only does the soft tower weigh much less, the shell type construction is considerably cheaper to fabricate on a cost per pound basis. Direct tower cost savings are substantial. Of perhaps even more importance is the fact that rotor stiffness and weight are not serious restraints when using the soft tower, permitting the use of heavy but economical and reliable rotor designs.

Though time does not permit a detailed review of every MOD-2 feature, the following is a list and brief comment on those other features most responsible for achieving the relatively low MOD-2 cost-of-electricity:

Drive Train Quill Shaft - As illustrated in Figure 23, the on line shaft frequency of approximately .5 per revolution economically attenuates the two per rev. alternating torques that are particularly troublesome with a teetered-tip control rotor configuration.

Tip Control - A feature that substantially reduces rotor weight and cost with only minor compromises in power output, startup and shutdown control, and torque oscillation.

Teetered Hub - First looked at primarily as a means of reducing rotor fatigue, the major payoff of this feature is a reduction in weight and cost of the nacelle, low speed shaft, yaw system, and tower.

Compact Planetary Gear Box - Selection and development of this advanced design gear box has resulted in over 100,000# system weight saving, a much simplified nacelle installation, and direct cost saving.

Upwind Rotor - The upwind rotor configuration slightly reduced rotor fatigue and resulted in a 2 1/2% increase in annual power produced while adding negligible cost to the yaw system. Impact on the yaw system is minimized with the teetered rotor.

Nacelle Located Microprocessor - The change from a ground located computer with a multiplexer in the nacelle to a microprocessor located in the nacelle resulted in both direct cost savings and a substantial reduction in anticipated maintenance cost.

Gin Pole & Hoist Erection & Maintenance - Very large wind turbines can experience severe maintenance costs as well as loss of power produced when held up for the expensive and sometimes unavailable large cranes required for major component replacement or repair. The MOD-2 solution is to provide permanent gin pole, hoist, and guy line foundations at each site, permitting the use of a relatively inexpensive gin pole and hoists. A secondary fallout of this basic maintenance provision is a convenient and economical means of system erection.

#### MAJOR PROBLEMS

I have been asked to report on major problems. At this writing, I am happy to report that except for the everpresent problems of schedule and budget, we are aware of no serious technical problems. But don't misunderstand; we anticipate problems will arise in subsequent program stages. However, at this point, we would have to call them unk -unks.

#### CONCLUSIONS & RECOMMENDATIONS

After working on the MOD-2 program for nearly two years, one conclusion is evident: Wind Power has come of age. It not only promises to be more practical than any of the other so-called alternate energy sources, but it is actually competitive with today's energy sources in many geographical areas.

The intent of the MOD-2 program has been to incorporate all concepts that show reasonable promise and, to the extent program scope has permitted, the intent has been implemented. Additional study of such potential advanced features as a fixed pitch rotor can no doubt be justified. However, we see the largest system improvement potential in a component-by-component study effort, applying value engineering disciplines as well as seeking efficiency gains. As has been proven true on our commercial aircraft programs, these improvements can best be made utilizing experience gained from a sizeable number of commercially deployed units.

MOD-2 can and will be improved with time, just as fifty years from now the then current systems can and will be improved. But using today's technology, no concept changes show sufficient promise to warrant any further delay in production deployment of wind power. Let's get on with it!

#### REFERENCE

1. Couch, J. P., MOD-2 Wind Generator Program, Third Wind Energy Workshop, Sept., 1977.

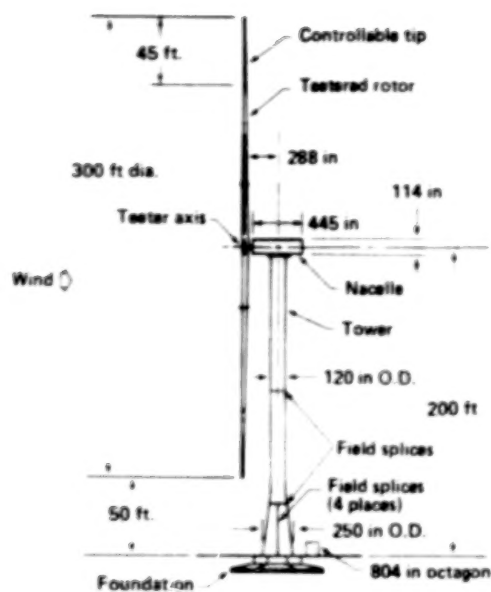


## DISCUSSION

- Q. How many planets do you have in the first stage of the gearbox?
- A. Actually we are developing two gearboxes for Mod-2. The primary gearbox has eight planets in the first stage; the alternate gearbox has six.
- Q. In your discussion of the tower loads, vibrations seemed to be linked to rotor dynamics. Have you determined what effect isotropic turbulence or even micro turbulence might have on tower loads? Also what limiting turbulent conditions did you consider?
- A. Oscillations due to both vortex shedding and turbulence have been considered in the Mod-2 tower loads using coupled modes analysis. The resulting response to vortex shedding was small. Tower loads due to the maximum statistical isotropic turbulence acting on both the tower and the rotor was also analyzed. We found that the maximum turbulence induced loads were less critical than steady extreme wind loads.
- Q. In your cost of energy equation, how do you handle the effect of inflation over the 30-year life?
- A. The cost-of-energy equation shown in my presentation was given to us by NASA as a Mod-2 program ground rule. However, we have looked at 30-year levelizing using a factor applied to the operation and maintenance term. Since operation and maintenance are a comparatively small part of annual cost, application of this levelizing factor has only minor impact on Mod-2 cost of electricity.
- Q. Your design is facing into the wind. How severe is the extreme wind load case, and what extreme wind velocity have you designed to?
- A. We had designed to an extreme wind of 120 mph. This case designs a very minor portion of the rotor and a major portion of the tower and foundation.

*Table 1. MOD-2 Design Requirements*

<u>Requirement</u>	<u>Value</u>
General	
Service life	30 years
Rotor orientation	Horizontal axis
Rotor diameter	300 feet
Environmental:	
Mean yearly wind speed	14 mph at 30 feet
Wind gradient	Variable power law
Wind speed duration	Weibull distribution
Altitude	0 - 7,000 feet
Lightning	Per NASA model
Seismic - Wind Turbine	Zone 3
Seismic - Foundation	Zone 2
Temperature range	40°F to 105°F
Rain, hail, snow, etc.	Yes
Max design wind	120 mph at 30 feet
Operation and maintenance:	
Fail safe unattended operation	Yes
Fire and ice detection	Yes
Network and turbine protection	Yes
Obstruction marking and lighting	Yes
Maintenance tools and vehicles	Commercial



Rated power	2,500 KW
Rotor diameter	300 ft
Rotor type	Teetered - tip control
Rotor orientation	Uprwind
Rotor airfoil	NACA 230XX
Rated wind @ hub	27.5 mph
Cut-off wind speed @ hub	45 mph
Rotor tip speed	275 ft/sec
Rotor rpm	17.5
Generator rpm	1,800
Generator type	Synchronous
Gear box	Compact planetary gear
Hub height	200 ft
Tower	Soft-shell type
Pitch control	Hydraulic
Yaw control	Hydraulic
Electronic control	Microprocessor
System power coefficient	0.382

Figure 1. General Configuration & Features

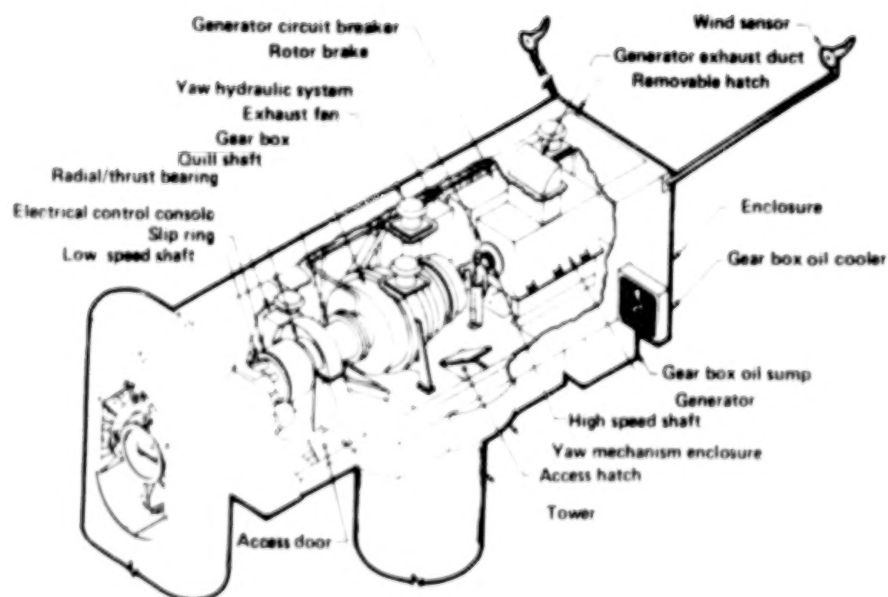
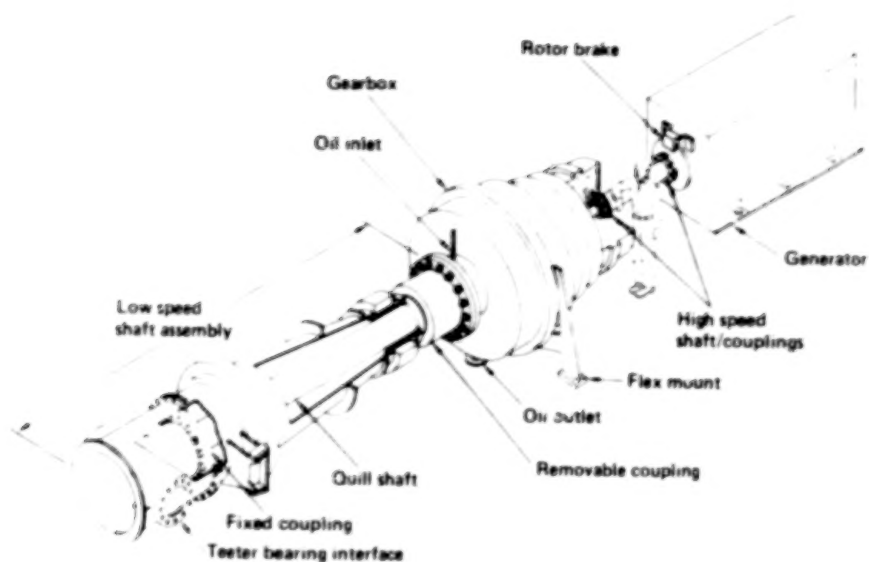
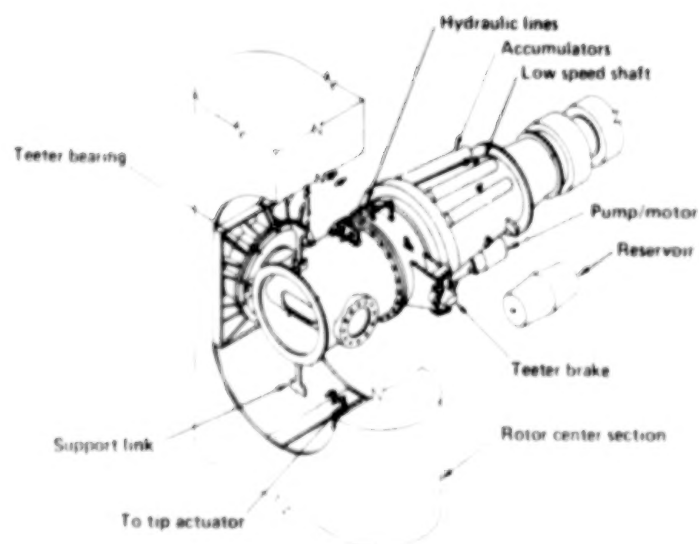


Figure 2. General Nacelle Arrangement MOD-2-107



*Figure 3. Drive Train*



*Figure 4. Pitch Hydraulic System Low Speed Shaft*

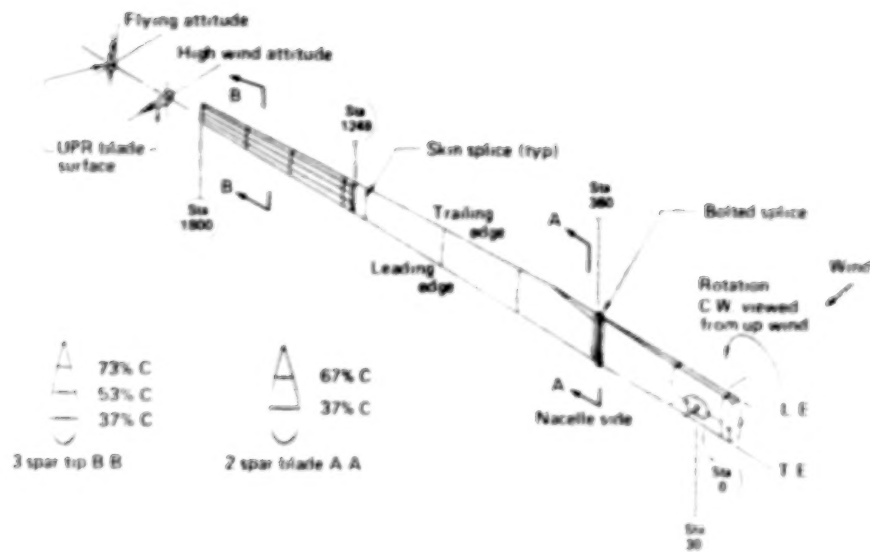


Figure 5. Steel Rotor Blade Configuration MOD-2-107

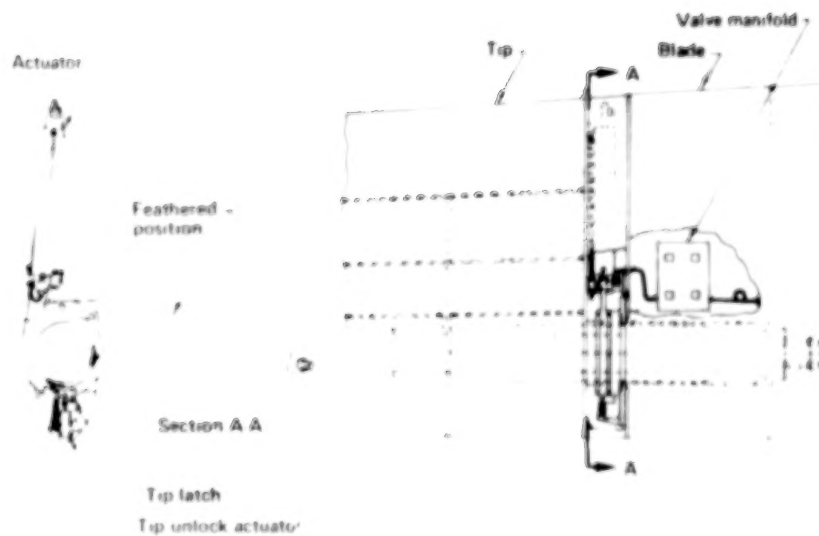


Figure 6. Pitch Control Mechanism MOD-2-107

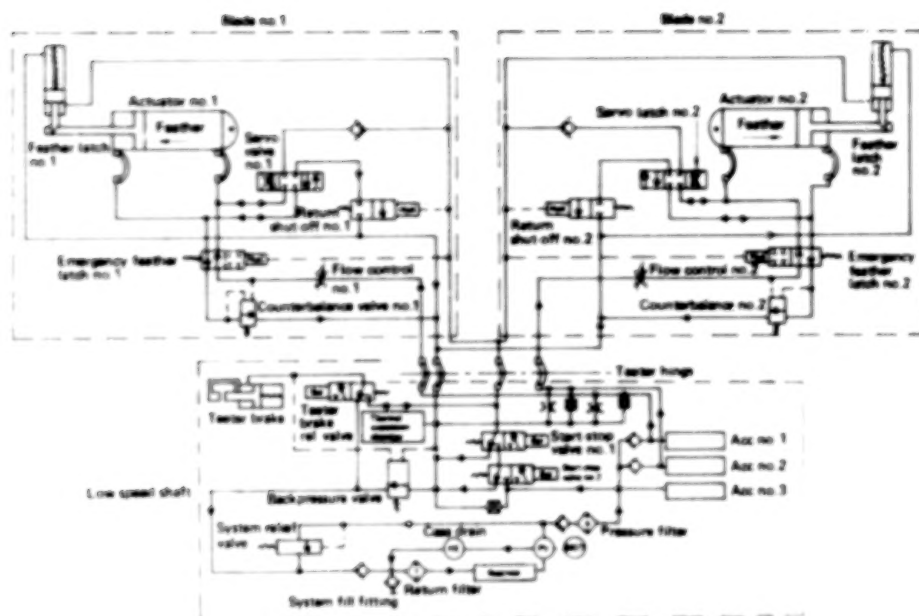


Figure 7. Hydraulic Schematic Pitch Control System MOD-2-107

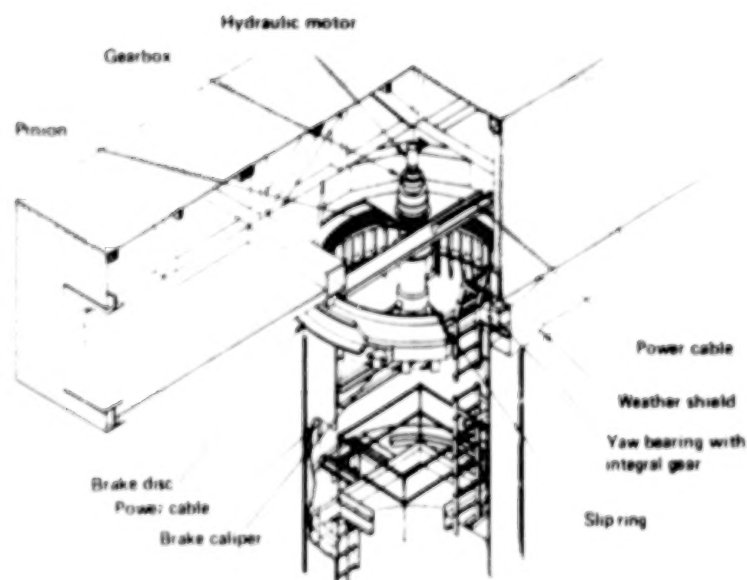


Figure 8. Yaw Drive Installation

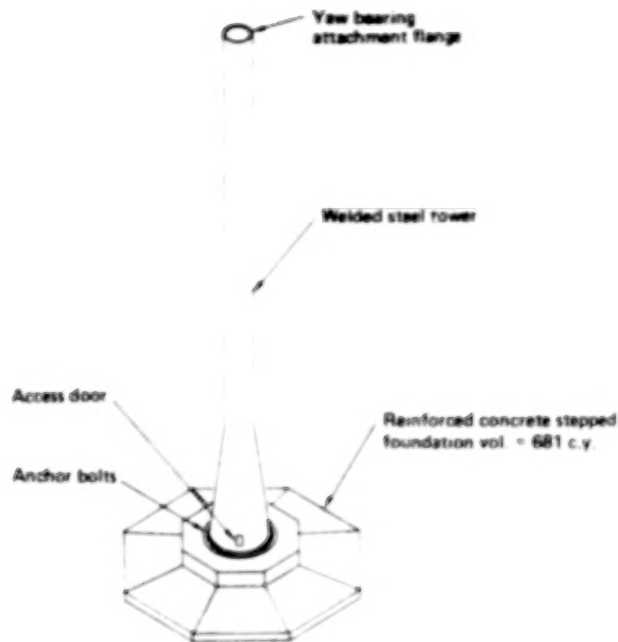


Figure 9. Tower/Foundation MOD-2-107

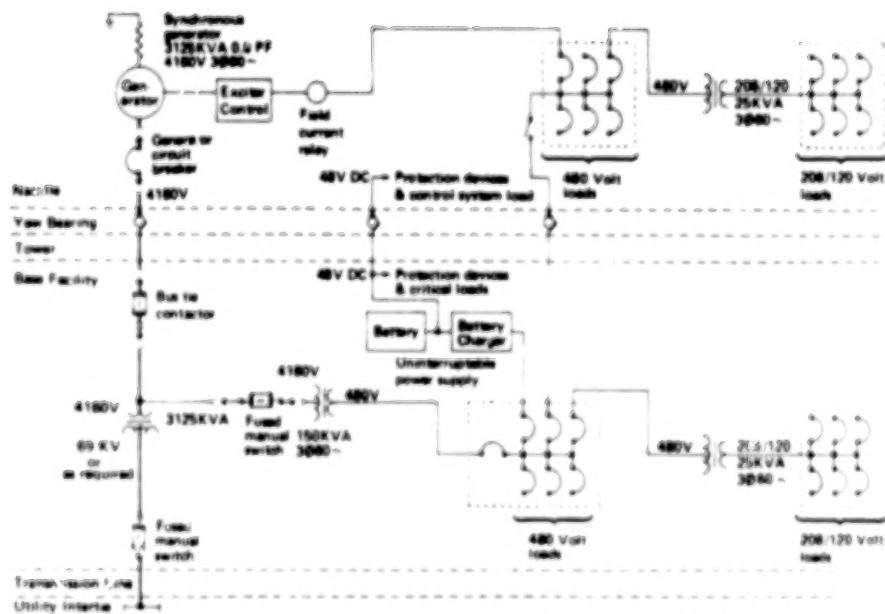


Figure 10. Electrical Power System - Power Network MOD-2-107



Figure 11. Control System Interface Diagram

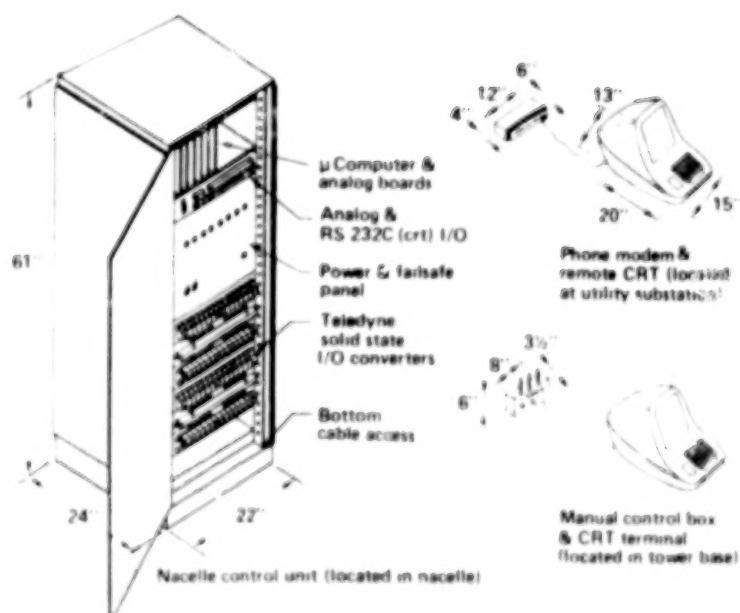


Figure 12. Control System Major Components MOD-2-107

Element	Weight (lbs)
Rotor assembly	169,567
Gearbox	39,000
Generator	17,000
Drive train components	39,892
Nacelle structure	40,832
Yaw drive	17,742
Misc. Nacelle equipment	4,705
Tower assembly	251,486
	<hr/> 580,204

Figure 13. Weight Summary

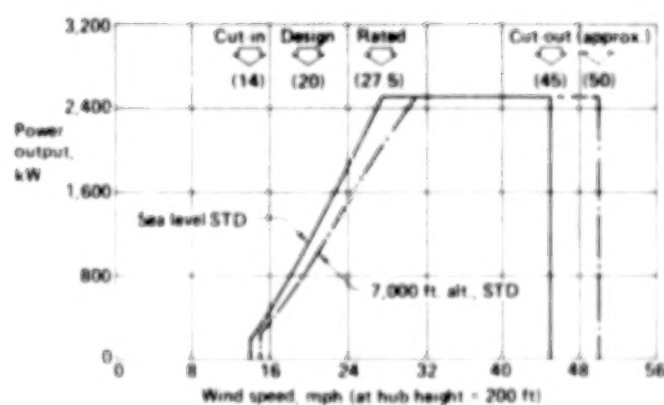


Figure 14. Power Output Vs. Wind Speed



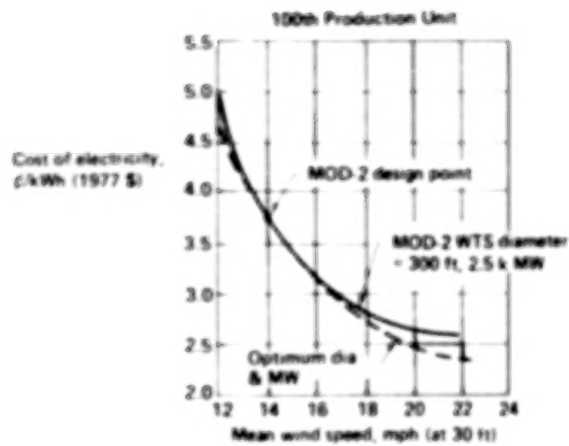


Figure 15. Effect of Mean Wind Speed on Economic Performance

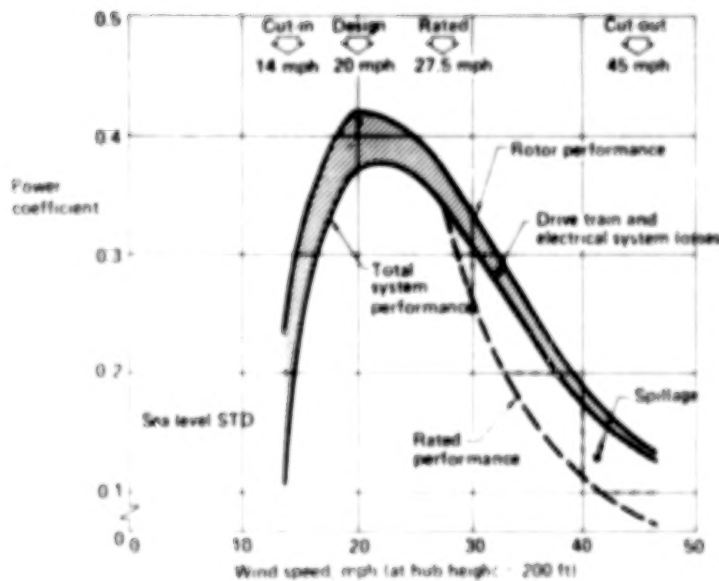


Figure 16. System Performance

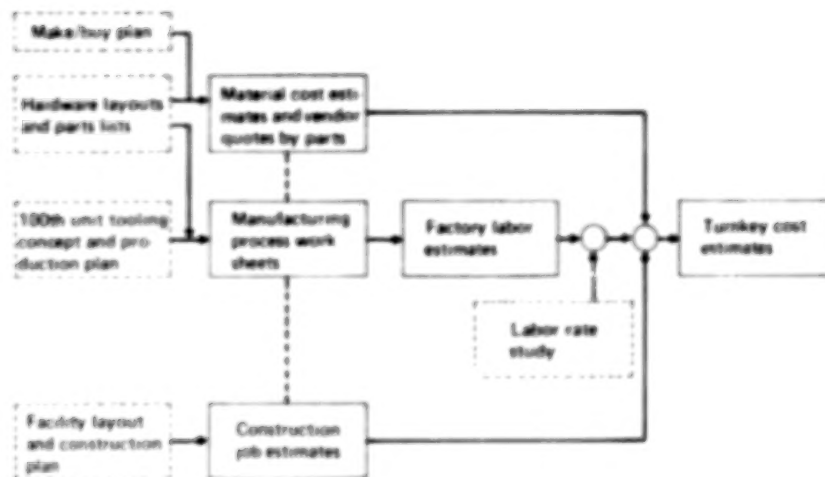


Figure 17. Cost Approach

Turnkey account	Cost
1.0 Site preparation	\$162,000
2.0 Transportation	29,000
3.0 Erection	137,000
4.0 Drive train	329,000
5.0 Rotor	379,000
6.0 Nacelle	184,000
7.0 Tower	271,000
8.0 Initial spares	35,000
8.A. Non-recurring	35,000
9.J Total initial cost	\$1,561,000
Fee (10%)	156,000
Total turnkey	\$1,717,000
10.0 Annual operations and maintenance	\$ 15,000

The cost estimating ground rules are as follows:

- All costs are in mid 1977 dollars
- Costs of installation and operation are based on a 25 unit farm
- Transportation costs are based on rail and truck transport over a distance of 7,000 miles

Figure 18. 100th Unit Production Costs

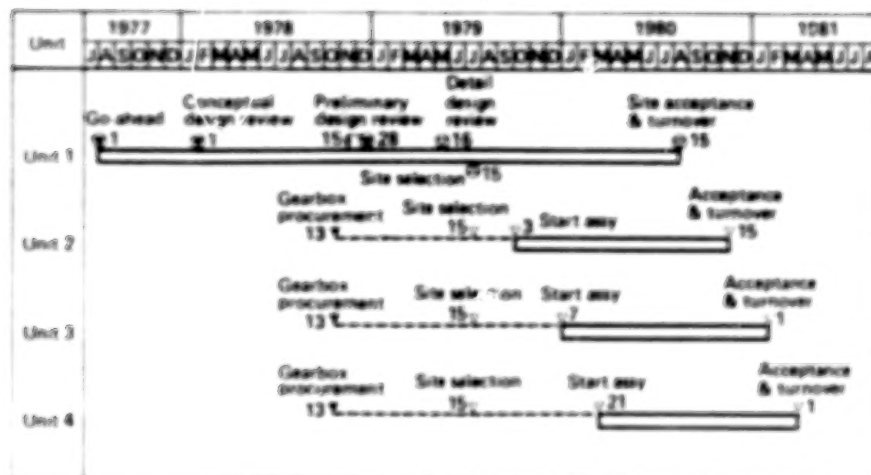


Figure 19. WTS MOD-2 Tier I Master Schedule

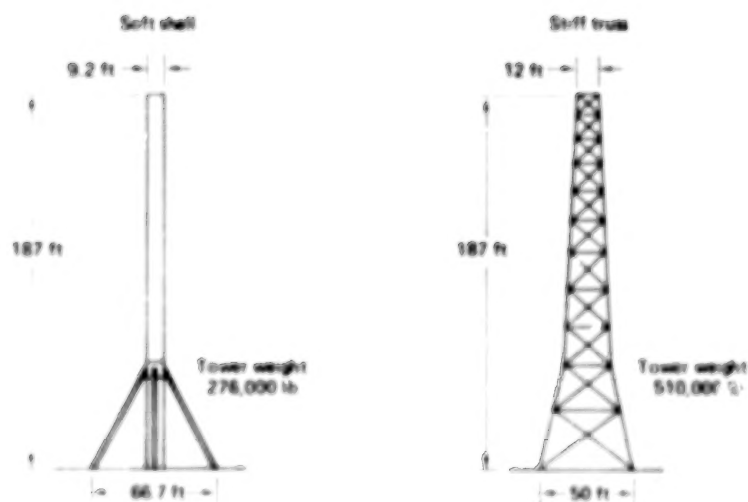


Figure 20. Tower Configurations

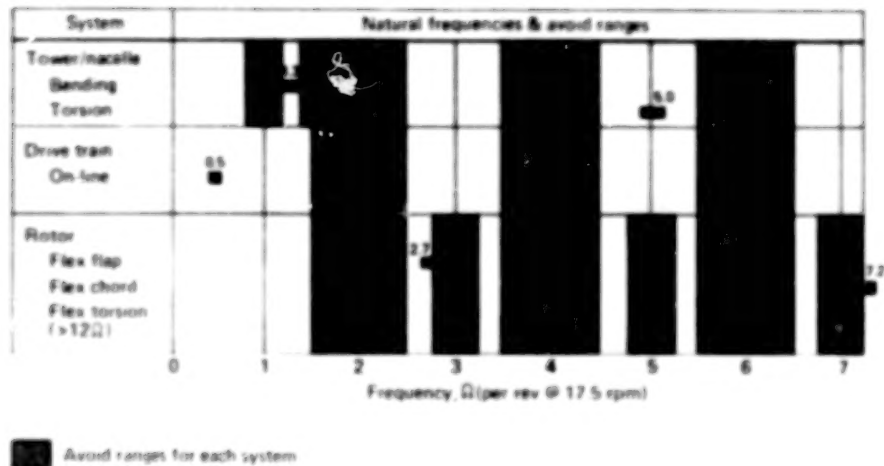


Figure 21. MOD-2 System Frequencies

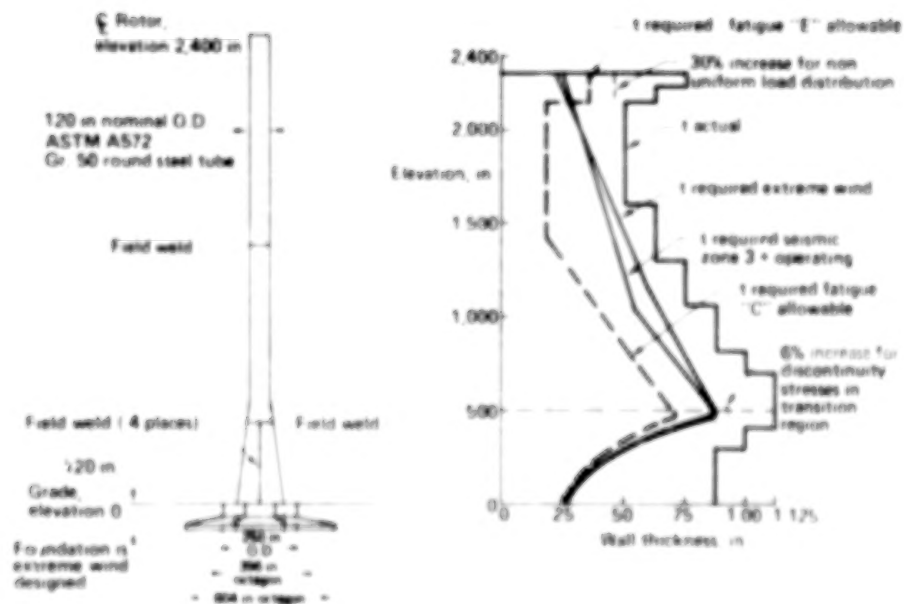


Figure 22. Tower/Foundation MOD-2-107

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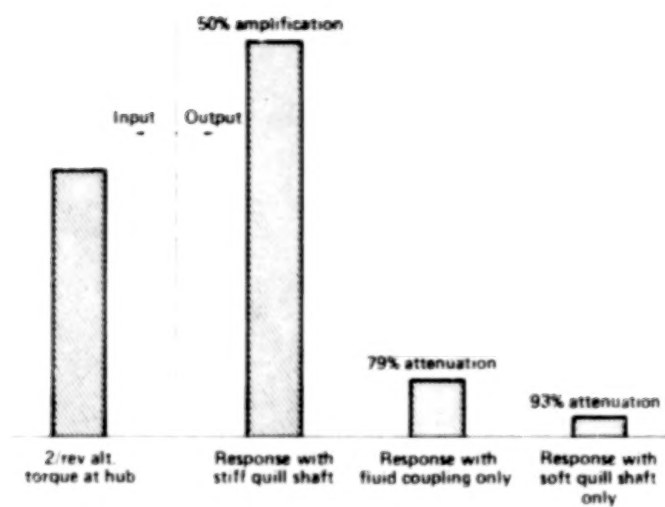


Figure 23. Response to 2/Rev Torsional Forcing Functions



WTG ENERGY SYSTEMS' MP1-200 -  
200 KILOWATT WIND TURBINE GENERATOR

Allen P. Spaulding, Jr.  
WTG Energy Systems, Inc.  
Buffalo, New York 14203

### INTRODUCTION

The areas to be discussed in this presentation are related to the preliminary design criteria as utilized on the MP1-200. The significance of these design criteria are based on the fact that the MP1-200 is the only wind turbine in operation today that is producing synchronous alternating current using a fixed pitch rotor configuration (fig. 1).

The MP1-200 is installed on Cuttyhunk Island, Massachusetts as part of the Island's independent utility grid system. The municipal utility on Cuttyhunk Island is diesel engine powered with an installed capacity of 465 kW. The annual demand curve, Figure 2, is plotted against the wind turbine's production rate. It is evident from this graph that the wind turbine will have a profound effect on the rate of fuel consumption with the exception of the demand peaks experienced during the summer months. Cuttyhunk Island was chosen as the site for the prototype because it was felt that this type of application was typical of the ideal installation for this type of generating system. As Figure 2 indicates, the wind turbine operates as the dominant power source much of the time. For a wind generator to operate effectively in this type of network it must be capable of maintaining its frequency while operating in parallel with any size conventional power plant.

The MP1-200 became operational as a test unit in June, 1977. Since that time it has been subjected to numerous operational and environmental tests. The machine has been run for extended periods at 70% over rated speed (80rpm) without damage to any components. It withstood wind velocities in excess of 100 mph on four separate occasions. This machine has proven the concept that a fixed pitch rotor configuration can be utilized effectively at a competitive cost to produce synchronous power under all operating conditions normally associated with conventional generating plants.

### PRELIMINARY DESIGN CONSIDERATIONS

The first step taken in the preliminary design process was to define the potential market for a 200 kilowatt wind generating system. The methods used were simple, direct canvassing of

potential end users; utility companies and large industry, located in areas with high wind regimes. The results of this survey indicated that potential customers would consider a system such as the MP1-200 at an installed cost of approximately \$1,000 per kilowatt. In addition the survey indicated that potential buyers wanted a design life for the major components of at least 20 years, synchronous power production directly from the system's generator, minimum service and maintenance, and completely unattended operation over the full operating range of the system.

The second step in the preliminary design process was to conduct an historical survey of large wind turbines. Of particular interest were the fatigue life of large systems and the method(s) used to control rotor speed. Of equal interest was the cost breakdown of these systems. The results of this survey are summarized below.

1. The Gedser Mill as constructed in Denmark in the mid 1950's appeared to be the most fatigue resistant large wind turbine built to date (1974). In addition the costs involved in the construction of this machine were within the guide lines that we had originally set.
2. We found no control systems in the large wind turbines of the past that could meet present day control requirements for synchronous generating systems. All wind turbine control systems up to that time utilized a variable pitch rotor as the primary speed control. The only way that these systems can produce constant voltage and frequency to meet present day standards of accuracy is by parallel operation with a much larger capacity grid system.
3. A third major problem that has plagued large wind turbines is rotor fatigue caused, primarily, by the in plane gravitational loads during operation. A second major contributor to rotor fatigue results from the location of the rotor down wind from the tower causing the blade to 'unload' once per revolution. In addition, this phenomenon has been found to cause erratic behavior in the generator's frequency control system due to momentary loss of torque at the rotor.

#### MP1-200 DESIGN CRITERIA

- All steel construction
- 30 year design life for all major hardware components
- Fixed pitch rotor configuration
- Rotor operation up wind of the tower
- Solid state control and speed governing

- Automatic, unattended operation
- Remote monitoring and control capability
- Operational range: 60deg. N to 50deg. S. latitude
- AC synchronous power produced directly from the wind turbine's generator, through out the systems' operating range, in either parallel or independent operation
- +/- 1% control accuracy of frequency and voltage in either parallel or independent generator operation
- \$1,000 per installed kW (1975 dollars)

#### MP1-200 WIND TURBINE DESCRIPTION

The MP1-200 wind turbine installed on Cuttyhunk Island, Massachusetts, utilizes a three bladed, 80 feet in diameter rotor operating upwind of the tower. The machine is constructed entirely of steel. The tower height, measured from ground level to the rotor's center line, is 80 feet. The rotor operates at a constant 30 rpm driving a 250 KVA synchronous generator through a 40:1 gear transmission. Blade tips rotate 60 degrees out of plane to provide aerodynamic braking. A 24 inch disc brake mounted on the high speed shaft is used for "parking" the rotor. Yaw position is controlled by dual hydraulic servo motors working through two speed reducing transmissions. The entire nacelle assembly rotates on a 59 inch platter bearing. A 72 inch disc brake is provided for locking the yaw position. The tower used is a pinned-truss type, constructed of Cor-Ten steel. Wind speed and direction are sensed on a remote tower and are used to control startup, shutdown and yaw sequences. Components are shown in figure 3.

A system for controlling the speed of the wind turbine has been developed by WTG Energy Systems which utilizes load modulation with the fixed pitch rotor configuration. An industrial process controller is used for the control and monitoring on the MP1-200. This processor represents an ideal compromise in cost, input/output capabilities, processing speed and reliable operation in rough environmental conditions. A versatile software feedback control algorithm is provided and utilized in the speed control system. The controller is an "off-the-shelf" item with no required hardware modifications.

#### PERFORMANCE AND OPERATIONAL EXPERIENCE

The MP1-200 has produced power in excess of 300 kilowatts in winds of 3<sup>5</sup> miles per hour. Operation begins in wind speeds above 8 miles per hour and rated output is achieved at 28 miles per hour. The machine is shut down when the average wind exceeds 40 miles per hour. Power varies directly with

the rotor response to variations in wind velocity about the mean. Power regulation or stabilization of the output is not used in this design. Fluctuations in power approach 30 percent of nominal output in high gusty winds. As a result of the relatively low frequency response characteristics of the high inertia rotor, deviations of this magnitude should not present a problem to most utility networks.

The pitch of the rotor is adjusted initially to reach its peak power coefficient in winds of 18 miles per hour. In winds above this level the rotor goes into a stalling condition. This condition was found to be gentle and predictable; power continues to increase up to the peak and levels gradually. This phenomenon inherently limits the maximum level of power produced.

Figure 4 shows a strip chart of the wind generator regulating independently of any other source. Regulation of the generator's speed is very good in winds up to 25 miles per hour and tends to degrade slightly above this speed as a result of the high frequency gust components common with higher wind velocities. Worst case accuracy of plus or minus 0.75 hertz (generator output frequency) is specified for isochronous operation. This accuracy is, of course, improved when operating in synchronism with a stable source of equal or greater capacity.

Figure 5 depicts an actual strip chart of the wind generator's performance under synchronous operating conditions. After the speed is adjusted the main contactor is energized. As is standard practice, speed droop is provided on the diesel plant and is adjusted at 2 percent. The load will be divided proportionally to the generator's speed setting. When the wind generator is capable of carrying the entire town load it will do so at a nominal frequency of 60 hertz. The diesel plant will at this point be idling because of its droop setting. As the wind generator's capacity drops (because of a decrease in wind velocity) its speed will begin to fall and the diesel set will pick up the proportion of load dropped by the wind generator thus allowing the generators to maintain nominal frequency while dividing the load proportional to the input torque of the wind generator. System frequency could fall as low as 59.5 hertz when the diesel is fully loaded and the wind generator is idling. This condition would occur when the wind velocity is varying around 8 miles per hour, and the wind turbine would be taken "off line" to prevent excessive reverse power flow. When the output of the wind generator is greater than the Town's demand the remainder of its output is dissipated in the load bank.

Numerous tests and refinements have been made to achieve a high level of performance. Modifications on the basic design have been directed in the following areas:

1. Yaw system drive torque and bedplate-to-tower coupling

The original hydraulic motors used to yaw the machine proved

to be insufficient in terms of torque capabilities. They were replaced with motor/transmission units. A slight decrease in the yaw rate occurred, but the torque was increased to a level sufficient to drive the machine under any condition. The bedplate to tower coupling was provided with a 72 inch disc brake with three hydraulic calipers. This brake maintains a very stiff coupling at this critical union.

## 2. Control system bandwidth

The original signal conditioning and output actuating equipment was found to be insufficient in response, to accurately control the frequency of the wind turbine in wind velocities above 25 miles per hour. The wind generator was always stable in operation when "locked" in synchronism with a stable source of approximately 4 times its nominal capacity. Used in "Infinite Bus" applications, interface should present no difficulty with the system as presently configured. Work is being done to increase the effective range of operation for remote applications such as Cuttyhunk. The faster control system should be operational by May, 1979.

## COST

We are continually working to increase the performance and lower the cost of this system, without sacrificing reliability. At present production costs, Figure 6, are of prime interest. Our goal, as stated earlier is \$1,000 per installed kW. At this time WTG Energy Systems is quoting a price of \$226,000 FOB the plant, or approximately \$1,130.00 per kW. We have calculated that with a production run of 5 units the per unit cost could be reduced by 30%. This would equal an FOB cost of \$158,000 or \$791.00 per kilowatt.

## FUTURE R & D REQUIREMENTS AND SUGGESTIONS

The areas in the design and operation of the MP1-200 system in particular, and, wind turbine generator systems in general requiring additional research and development are listed below.

1. Increased field testing of large wind turbines interfaced with small hydro electric installations should be given high priority. This application has the potential of allocating greater capacity credits for both systems.
2. Increased emphasis on field testing wind turbine/diesel packaged systems. Emphasis should be placed on the design of diesel engine combustion requirements operating with reduced loads and the retrofitting of existing units for similar operational parameters.
3. Field testing of multiunit wind generating systems interfaced



with conventional grid systems. Of particular interest are combination systems each with equal installed capacities.

## CONCLUSION

To date the system has met or exceeded the original design criteria. We feel that this system demonstrates that synchronous power can be produced directly from a wind driven generating system at a cost that is competitive, in many areas, with conventionally powered generating systems.

We are continuing to work on improvements in the control system, on production techniques and methods of installation to further reduce the system's cost and increase its reliability.

## DISCUSSION

Q. Can you discuss the blade construction?

A. That will be covered in a later paper by Bob Barrows, the chief engineer of this project. I will let him answer that question.

Q. What is the rotor diameter and rated power?

A. The rotor diameter is 80 feet, and the rating is 200 kilowatts in a 26 to 28 mph wind.

Q. What is your assessment of the market potential, in dollars per year over the next five years, for intermediate size machines?

A. Since we had our press conference, we have written about ten proposals to utility companies all over the world--in Australia, Africa and some in the United States. There are about 500 small diesel utilities in the United States that are in high wind areas. That's the best I can tell you right now. We are in the process of doing a lot of work in this area. As a matter of fact, we are spending most of our money and time on this aspect of the business.

Q. How much energy is being discarded during the winter and summer months?

A. No energy is thrown away in the summer, as this is the island's peak demand period. At night during the winter, the demand for the island often drops to as low as 20 kilowatts while the wind turbine is operating at maximum output. During this period quite a bit of power is discarded.

The island's power plant is a municipal plant. Presently no heat, hot water and cooking requirements are part of the utility's demand. If more of the island's energy requirements were served by the utility, less would of course, be burned off. Ultimately, it will be up to the utility and the residents to decide how much of their total demand should be electrical and the economic value of the conversion.

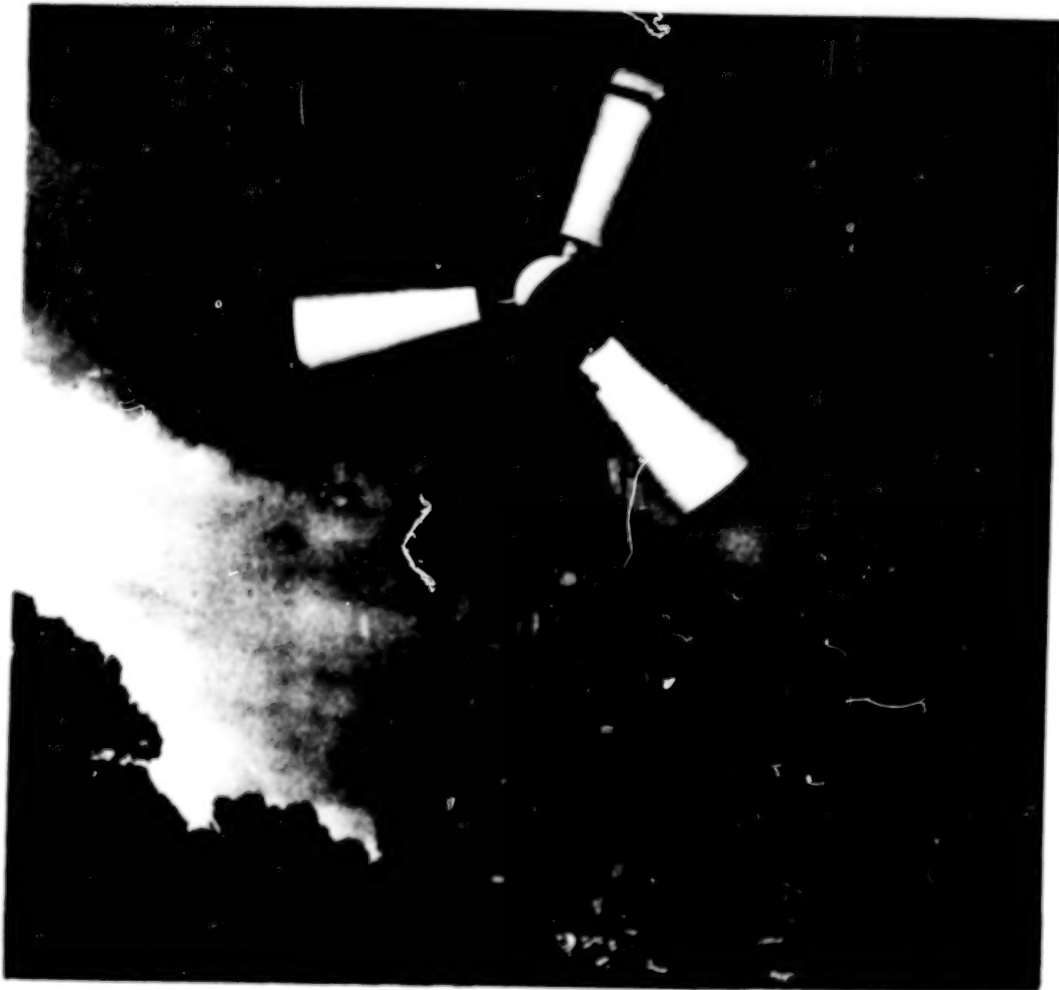


Figure 1. - MP1 - 200 wind turbine generator. (Photo courtesy of Eagle Signal Division, Gulf and Western Manufacturing Company.)

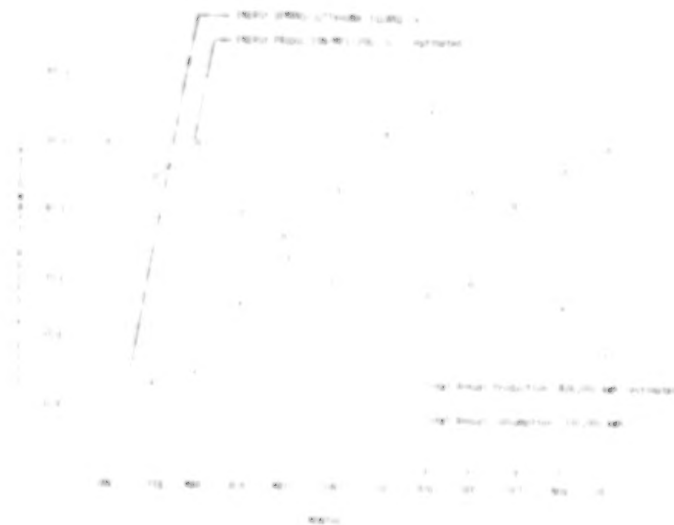


Figure 2. - Energy comparison.



Figure 3. - Wind turbine generator components.



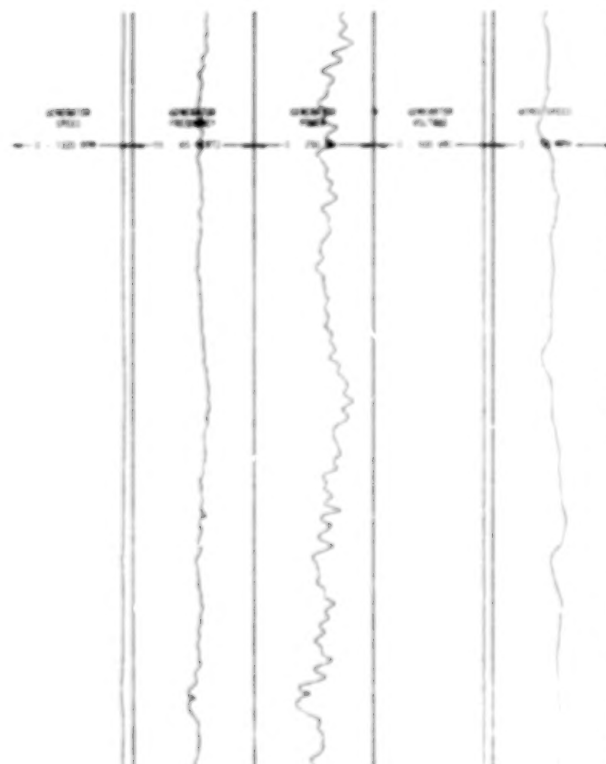


Figure 4. - Wind generator regulation.

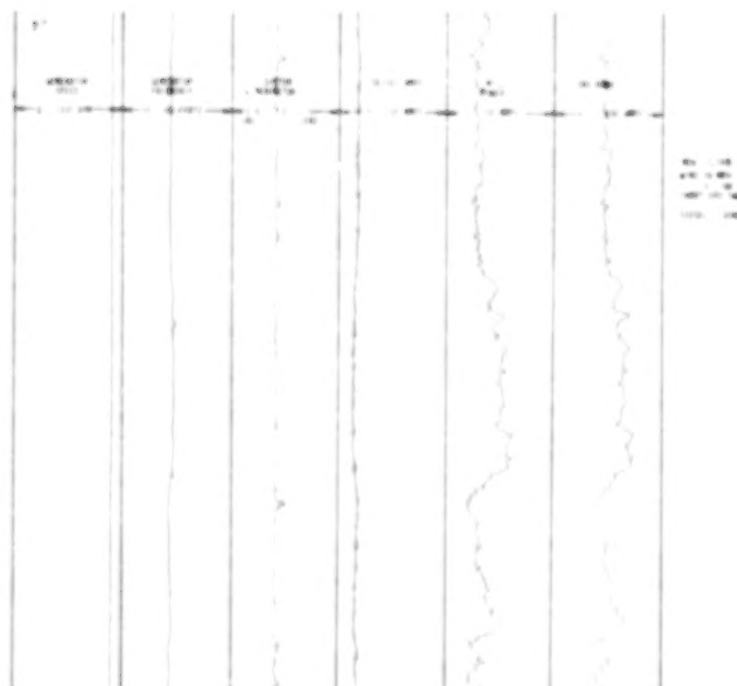


Figure 5. - Performance under synchronous operating conditions.

<u>COMPONENT</u>	<u>PER CENT OF TOTAL COST</u>	<u>COST</u>
RIBS	18.4%	\$ 7,560.00
SPARS	12.3%	5,040.00
HUB	18.4%	7,560.00
TIP FLAPS	13.8%	5,670.00
BLADE SKIN	5.4%	2,250.00
MISC. HARDWARE	4.9%	2,030.00
LABOR	26.8%	10,810.00
<u>TOTAL</u>	<u>100%</u>	<u>\$ 42,920.00</u>

Figure 6. - Cost breakdown MP-1 200 rotor system by major components (sale price for limited production).

## SPECIFICATION, SITING AND SELECTION OF LARGE WECS PROTOTYPES

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### Introduction and Research Unit update

The Swedish Wind Energy Programme was started in 1974 with preliminary feasibility studies. These indicated that wind power could become an economic reality in Sweden, and that the technical problems would not be unsurmountable. This led to a decision by NE in 1975 to design and install a Wind Power Research Unit to study the technical problems associated with wind power at a semi-scale level. The contract for this Unit - with main characteristics as given below - was given to Saab-Scania Co.

### Characteristics of Swedish Research Unit (Figure 1)

Tower:	Concrete, diameter 2 m
Hub height:	25 m
Hub type:	A) Rigid B) Flapping
Turbine diameter:	A) 18 m B) 24 m
Turbine rpm:	77
Rotor blades:	A) Aluminium B) GRP C) CRP+GRP
Rated power:	63 kW (75 kW)
Generator:	380 V, asynchronous
Grid voltage:	10 kV

The Unit was operative in April 1977, underwent delivery tests and debugging during 1977 and began giving test data for the aluminium blade/rigid hub combination late in 1977. That combination accumulated 846 hours of operation before the hub was changed in May 1978. The combination aluminium blade/flapping hub is now operative, accumulating about 1200 hours in early April 1977, total hours of operation now being above 2000. Rotor blades will be changed into a GRP-set in May 1979, and late in 1979 into a CRP+GRP-set with increased diameter (24 m).

In the first six weeks of 1979 the Unit was in remote controlled, routine "utility operation" with only weekly inspections. 400 hours were accumulated - as the winds blew - with only one snag: the temperature in the morning of January 29 was so low ( $-30^{\circ}\text{C}$ ) that the Unit refused to start because of  $-5^{\circ}\text{C}$  in the main bearing! The technical availability during the period was 97%.

### Prototype Specification Development

Continued systems analysis work, and the early experiences of the Research Unit was the basis for a decision by NE in late 1977 to develop a "Technical Specification for Design and Installation of Wind Turbine Systems in Sweden".

This specification was developed during October 1977 ~ April 1978, with some detail changes in September 1978. Our systems work had given the following rather clear indications:

- horizontal axis machines advantageous from most points-of-view,
- optimum turbine size in the range of 60-100 m diameter,
- hub height should be roughly equal to turbine diameter,
- concrete and steel towers roughly equal in feasibility and cost,
- blade materials and hub types should be tested in real life.

These and other deliberations led to the conclusion, that a functional Technical Specification should be written, to give a reasonably wide frame for proposals from prospective manufacturers. The frame boundaries should be given by reasonable physical restrictions, functional requirements and the electric supply network.

The Technical Specification was produced by a committee - disregarding the proverbial camel being a horse designed by a committee - chaired by the author of this paper. The committee included aerodynamics, structures and control systems consultants together with meteorologists, representatives of the two largest Swedish utilities, Vattenfall and Sydkraft, and furthermore development and engineering people from two prospective manufacturers, Saab-Scania and Karlskronavarvet.

Based on a general understanding within the committee concerning the functional approach and the indications from the systems analysis efforts, the work of the committee was organized as follows:

- the consultants were to draft all written material of the main specification, and to develop load cases and functional requirements,
- the meteorologists were to produce "best available" data concerning wind conditions (median winds, extreme winds, turbulence spectra) to be used in connection with the load cases and for performance calculations,
- the utilities were to define necessary electrical data and the requirements at the interface between WECS and grid, together with functional requirements for accessibility and maintainability,
- the prospective manufacturers were to give their comments and suggestions concerning the applicability of functional requirements and load cases, and also to develop recommendations concerning methods of calculation for certain problem areas, to be appended to the main specification.

In spite of the complexity, this scheme worked out quite weell during the few hectic months allocated for the job. Everyone engaged in this specification process took his task as a challenge, which is the only way to do it, when the task and its schedule seems impossible. The simple fact, that all those engaged knew each other from earlier projects, was probably a very helpful factor.

#### Summary of Prototype Specification

The final issue of "Technical Specification for Design and Installation of Wind Turbine Systems in Sweden" was published 1978-09-15. It has been distributed for information to all countries participating in the different international wind power projects of the International Energy Agency (IEA). The specification was written in English from the start, to facilitate international technological exchange.

#### Contents of Technical Specification

1. General
2. Definitions
3. Operational Conditions
4. General Requirements
5. Strength Requirements
6. Design, Construction, Erection
7. Instrumentation and Data Acquisition System
8. Inspection and Testing

Section "General" describes the purpose of the specification, states the encouragement of new concepts and innovations for the prototypes and the need for consideration of the visual appearence of the unit. It also states, that deviations from the specification are allowed only after negotiations with and approval by NE.

Section "Operational Conditions" gives the site wind characteristics - where we used data from Sturup Airport in southern Sweden as a common basis for the proposals, as the sites were not defined at the time. These characteristics consisted of:

- median wind velocity profile,
- wind duration during the year,
- extreme wind velocities with height profile,

- gust spectra with probability density and cross spectra definitions,
- local wind shear.

This section also deals with the general environment, access roads and transportation and the electrical network to be considered.

The "General Requirements" describe the main physical limitations and the required operational envelope, as in the table below:

#### Main Characteristics of Prototypes

Rated power (generator)	2-4 MW
Turbine diameter	70-90 m
Number of rotor blades	2-3
Inclination of rotor axis	Optional
Nominal tip speed	≤170 m/s
Minimum hub height	Equal to diameter
Generator system	Optional
Cut-in wind speed	6 m/s
Cut-out wind speed	≥21 m/s
Rated wind speed	Optional
Blade pitch control	Required
Remote control and monitoring	Required
Access to nacelle during ops.	Required

This section also describes the minimum functional modes, control system functions and the minimum functions of the electrical system of the unit.

The section concerning "Strength Requirements" contains definitions on load categories and load character, required factors of safety and probability of failure ( $\leq 10^{-5}$ ) during the service life (30 year). The load cases to be taken into account for structural design are furthermore defined, as summarized in table I.

Besides the definition of the different load cases, directives are given concerning the applicability of certain norms for erection loads and handling of heavy components. Furthermore some considerations on divergence and flutter speeds are given, to be above 36 m/s when in operation, and above 51 m/s when parked. A safety-factor of 1.5 regarding "toppling over" of the entire prototype on its foundation is prescribed.

The section on "Design, Construction, Erection" states that the technology used should be based on proven experience, and provide for future quantity production. It goes on to describe applicable design codes and standards, and then gives the general design considerations to apply to the different main components (wind turbine, machinery, nacelle, tower, control system and electrical installation with network connection). Blade airfoil and planform are optional. Machinery is optional, but the generator has to live with certain requirements as defined by the grid. The general functions of the electrical system are specified, and the main requirements

for its connection to the grid (30 kV and 50 kV respectively for the two sites).

The same section also deals with reliability and maintenance aspects, defining a design system life of 30 years and a minimum annual availability of 90% during the system life. Personnel safety is stressed and emergency evacuation from the nacelle is required. Consequences of component failure are to be analyzed by the contractor, the only strict requirement being, that a blade failure shall not dislocate or severely damage nacelle or tower. Lightning protection is specified according to a lightning model based on Swedish lightning statistics.

The "Instrumentation and Data Acquisition System" is only specified as to its main functions, and as to what data to be measured. The latter are divided into two groups: (1) power, energy and efficiency data; (2) engineering data. The former consist mainly of RPM, torque, active and reactive power, energy, voltage and frequency data in various points of the system together with wind data from a separate mast. The latter consist mainly of stress, temperature and vibration data, qualified by correlation to wind and power data, and by high resolution transient measurements.

"Inspection and Testing" is also defined in general terms, requiring contractor-developed plans for design control, factory tests, quality control, tests on site and acceptance tests. Among the required tests to be performed are:

- simulated lightning tests in case of non-conductive rotor blade material,
- measurement of stresses at several critical points of an entire rotor blade with limit loads applied,
- simulated function tests of various subsystems including all control loops before erection of the unit,
- ground resonance test of blade and of the entire turbine and nacelle on its turntable before erection,
- ground resonance test of tower at site.

The general schedule for the various activities of the Inspection and Testing process is described in Figure 2.

#### Request for Proposal

In April 1978 a Request for Proposal was mailed to those Swedish companies that had showed a serious interest in developing large-scale wind power prototypes. Such a request is an official document according to Swedish law, which means that any person or company can study the RFP at NE and respond to it. However, as this procurement of wind power prototypes is what is termed a "Negotiated Procurement", NE only has to consider the invited bidders.



The RFP was sent to six companies, of which two joined forces within short, resulting in five proposals from the following groups:

- Götaverken Motor AB (part of the State Shipyard group) Gothenburg;
- Karlskronvarvet AB (part of the State Shipyard group) Karlskrona, together with Hamilton Standard;
- Karlstads Mekaniska Werkstad AB (KMW, part of the Johnson group) Kristinehamn, together with ERNO, Bremen;
- Kockums Varv AB (part of the State Shipyard group) in Malmö, together with MAN, Munich;
- Swedewind (consortium of Saab-Scania AB and Stal-Laval AB) in Linköping.

The RFP consisted of a document stating the Conditions of Tender plus the Technical Specification as described above, together with various technical background material to give as comprehensive as possible common technical basis for the five bidders. The Conditions of Tender stated - among other things - that each invited bidder would be paid the sum of 1 million Sw.Kr (\$ 230 000) for his design study as part of his proposal.

In September 1978 the Technical Specification was amended in some details - as agreed with the bidding companies - and a Draft Contract for the procurement was issued, the latter only to serve as a guideline for later negotiations.

Proposals, containing fairly elaborate design studies, were received from the five bidding companies at the given deadline October 31, 1978.

#### Siting of Prototypes

The siting process was started already in February 1978 with the formation of a siting committee, chaired by the author of this paper as representing NE and composed of representatives for the County Governments of Malmöhus, Gotland and Uppsala Counties and for the two utilities that will operate the prototypes, Vattenfall (State Power Board) and Sydkraft (South Sweden Power Co).

The Siting Committee had to consider the following main factors in the process:

- wind conditions
- terrain and ground conditions
- nature conservation limitations
- environment and safety
- local planning and building regulations.



The committee had to formulate a recommendation for the final siting and to work out a basis for the final siting decisions to be taken by the following bodies:

- NE: technical siting
- County Government: conservation and environment
- Community Council: planning and building permit.

The most important factor to be considered was the wind conditions. Based on contour maps of Sweden with median winds at 50 and 100 metres ASL a decision on the general areas of interest could be taken. These were:

- southwestern Sweden in the province of Skåne
- the island of Gotland in the Baltic
- the Baltic coast of northern province of Uppland.

A visual inspection of these areas, coupled with local know-how of wind conditions, and taking terrain, forested areas etc into account, narrowed the choice to 8 small areas of about 2 sq.km. each. As other priorities were given for Skåne and Uppland, we could plan our final wind assessment for only these areas. The methods used for this assessment were the following:

- free pilot ballons measured by theodolites
- stability checks with SODAR (Gotland only)
- high mast checkpoint (Gotland only).

In spite of a less windy autumn than usual - as you would expect when you really want some wind - and fairly cold weather beginning in November 1978, the wind assessment worked out quite well during September-December 1978. The measured data was treated by a special computer program to increase accuracy by statistical methods. The conclusions were:

- the isovent maps were generally correct.
- different sites on the southern coast of Skåne were very similar.
- the assessed sites on Gotland were rather different with some unexpectedly large roughness effects but a "best site" could easily be found.

The Siting Committee recommended to NE - and NE duly decided likewise - to site one prototype at Maglarp in the province of Skåne, south of the city of Malmö, and one prototype at Näsudden on the island of Gotland. These sites are shown on Figure 3.

### Selection of prototypes

After receiving the prototype proposals, the selection process was started. Once again a committee was formed for the technical evaluation and selection process. This committee was almost the same as the one writing the specification, except - of course - that no prospective manufacturers were present. On the other hand, the work performed by the participating utilities was increased considerably, as they started to look deep into operation and maintenance aspects of the proposals.

We formulated a system of evaluation criteria - or perhaps rather evaluation aspects - breaking down the design concepts of the different proposals into successively finer details. The scope of this evaluation method was defined as to form a basis for:

- uniform evaluation of proposals
- objective judgement of technical problems
- distribution of work within the committee
- checking off the completeness of evaluation.

The evaluation aspects were divided into four groups, as listed below with the main contents of these groups.

### System Design

Was evaluated for the prototype and for the design implications for future series deliveries. The following subsystems of the prototype were studied:

- wind turbine (rotor and hub)
- machinery and nacelle
- tower and foundations
- control and servo systems
- electrical installation
- safety and maintenance equipment
- system integration

The aerodynamics, system dynamics and load characteristics were studied.

Operational and maintenance feasibility was evaluated.

### Performance

Was studied from purely technical and from operational viewpoints.

- wind/power conversion efficiency
- machinery losses
- operational availability
- failure-mode consequences
- system life estimates
- personnel safety

Cost-benefit analysis was also applied, partly for the prototype functions, but mainly for the series cost versus energy production situation.

#### Prototype delivery

The completeness and scope of the proposed delivery was compared with requirements.

The time and capacity planning for the realization of the prototype delivery was checked against independent project planning methods.

Contractual conditions as presented by the bidder were noted when differing from NE requirements. These questions are brought up in the final negotiations with the bidders.

The suitability for evaluation of the proposed design concept, was discussed in comparison between all five proposals, in order to arrive at a "mix" of design concepts in the final selection, that will give us a good technical coverage of what we consider to be the main development problems. More about that will follow later in this paper.

#### Contractor credibility

This part of the evaluation process was not considered critical, as all bidders are highly serious companies. Known differences, mainly in technical resources and know-how, between the five bidders were listed, to be used in the final comparisons.

When all these aspects were broken down into detailed technical "problem points", the committee worked its way straight through all proposals, judging the design solutions, calculations of loads and stresses, performance, planning etc with a very simple scoring system:

- 0 = not supplied, not dealt with or insufficient
- 1 = acceptable from all viewpoints
- 2 = more than required or special advantage.

We did not weigh the different aspects against each other, but merely summed up all the scores to arrive at a preliminary technical conclusion.

The method proved to give very conclusive results, we really never were in great doubt about our judgements.

After more detailed investigations concerning performance and stress calculations and cost-benefit aspects, we had to revise some of the given scores. From that point, the committee had to develop its own philosophy concerning the technological span of the two-prototype program, to arrive at a reasonably safe basis for the technical and economical recommendations on future wind power in Sweden, which are the target for the prototype testing program.

The basic reason for choosing a prototype program with more than one unit - we had originally planned for three units - was that our systems analysis projects had pointed at the necessity to evaluate and test more than one design concept. We were convinced, that we would otherwise not be able to predict with any certainty the future pro's and con's of wind power.

Within the general limits of fairly large, horizontal axis machines, there are still many options, such as:

- upwind or downwind turbine
- number of rotor blades
- rotor blade material
- type of hub
- synchronous or induction generator
- controlled or free in yaw
- rigid or soft turbine-tower dynamics
- tower material

We will give emphasis to selecting and testing the following conceptual differences:

- steel or concrete towers
- metal or composite rotor blades
- two different hub types
- soft or rigid towers
- synchronous or induction generators

These priorities concerning the "technological span" to be tested were used as "weighting factors" for the scores given under the various evaluation aspects. An assessment of know-how in the form of systems analysis background and methods for the different bidders was also used as such a factor.

This has led us up to a very definite conclusion as to which proposals we would like to buy from the technical viewpoint. Present negotiations with the bidders concerning prices, schedules and other more commercial conditions will show if the technical conclusions will be upheld also in the cold light of available money.

Our general time schedule for the continued prototype program calls for:

- |   |              |
|---|--------------|
| - Contracts signed                      | June 1979    |
| - Meteorological mast installed         | October 1979 |
| - Design phase ended                    | March 1980   |
| - Manufacturing ended                   | June 1981    |
| - Tower erected at site                 | March 1981   |
| - Installations ended, unit operational | Late 1981    |
| - Delivery tests completed              | Early 1982   |

At the "4th Biennial Conference and Workshop on WECS" in Washington D.C. in October this year, we hope to be able to present the selected prototypes in more detail, presumably by the happy Contractors.

#### Discussion

Q. Are all of your potential contractors Swedish organizations?

A. The main contractors are Swedish, and they have foreign partners. One of them is Hamilton Standard and two are German contractors.

Table I . Load Cases for Prototypes

- Case 1: Normal operation in steady winds.  
10 minute mean winds between cut-in and cut-out speeds.
- Case 2: Superimposed periodic and stochastic loads.  
Periodic fatigue loads (wind profile, tower shadow, gravity forces). Stochastic turbulence loads.
- Case 3: Sharp gradient wind shear.  
At  $V_R$  and normal RPM a vertical wind shear of 0.2 m/s/m.
- Case 4: Blade angle faults in normal operation.  
Possible control malfunctions and their consequences to be analyzed.  
A: Wind is at cut-out speed. RPM is nominal. Blade pitch ( $\beta$ ) instantaneously set at  $\beta = \beta(V_R)$ .  
B: Wind is at rated speed. RPM is nominal.  $\beta$  instantaneously set at  $\beta(\max)$ .
- Case 5: Wind turbine over-speed.  
Wind is at cut-out speed.  $\beta = \beta(V_{CO})$ . Torque reaction suddenly lost. Overspeed set by control system at  $RPM \leq 1.25 \times \text{nominal RPM}$ .
- Case 6: Loads on wind turbine in emergency braking.  
 $V = V_{CO}$ .  $RPM \leq 1.25 \times \text{nominal RPM}$ . Turbine being stopped by emergency braking system, as designed.
- Case 7: Loads due to electrical faults.  
Sudden cut-off (zero torque). Short circuit (dynamic oscillating torque).
- Case 8: Loads on parked prototype.  
Define parking geometry. Define yaw response.  
A: Symmetrical extreme gale wind.  
 $V = 51 \text{ m/s}$ .  $C_L = C_L(\max)$  over entire blade.  
B: Unsymmetrical extreme gale wind. (Applies only in case of vertical parking).  
 $V = 51 \text{ m/s}$ .  $C_L = C_L(\max)$  over entire upper blade,  $C_L = 0$  on entire lower blade.  
C: Parked with critical fault.  
Locked in yaw. Blade feathered vertically. Wind transversal to nacelle at  $V = 43 \text{ m/s}$ .  $C_D = 1.8$  over entire blade.
- Case 9: Ice loads.  
A: On parked prototype.  
Blades feathered in parked position.  $V = 43 \text{ m/s}$ . 50 mm ice on both sides of the entire blade, at  $\beta = 0.9$ . In horizontal parking  $C_L = -0.8$ .  
B: On prototype in operation.  
 $V = V_{CO}$ . Normal RPM and pitch angle. Leading edge ice buildup. Sudden loss of ice on one blade. Unbalance.
- Case 10: Bird collision with blade.  
 $V = 1.5 \times V_R$ ;  $V_{BIRD} = V + 15 \text{ m/s}$ . Bird weight 4 kgs. Bird impact at  $(0.7-1.0) \times R$  at or near leading edge. May not cause damage to the load carrying structure or cause sizeable parts to be thrown off.



Figure 1. Research unit with flapping hub.



Figure 2. General schedule for the inspection and testing process.

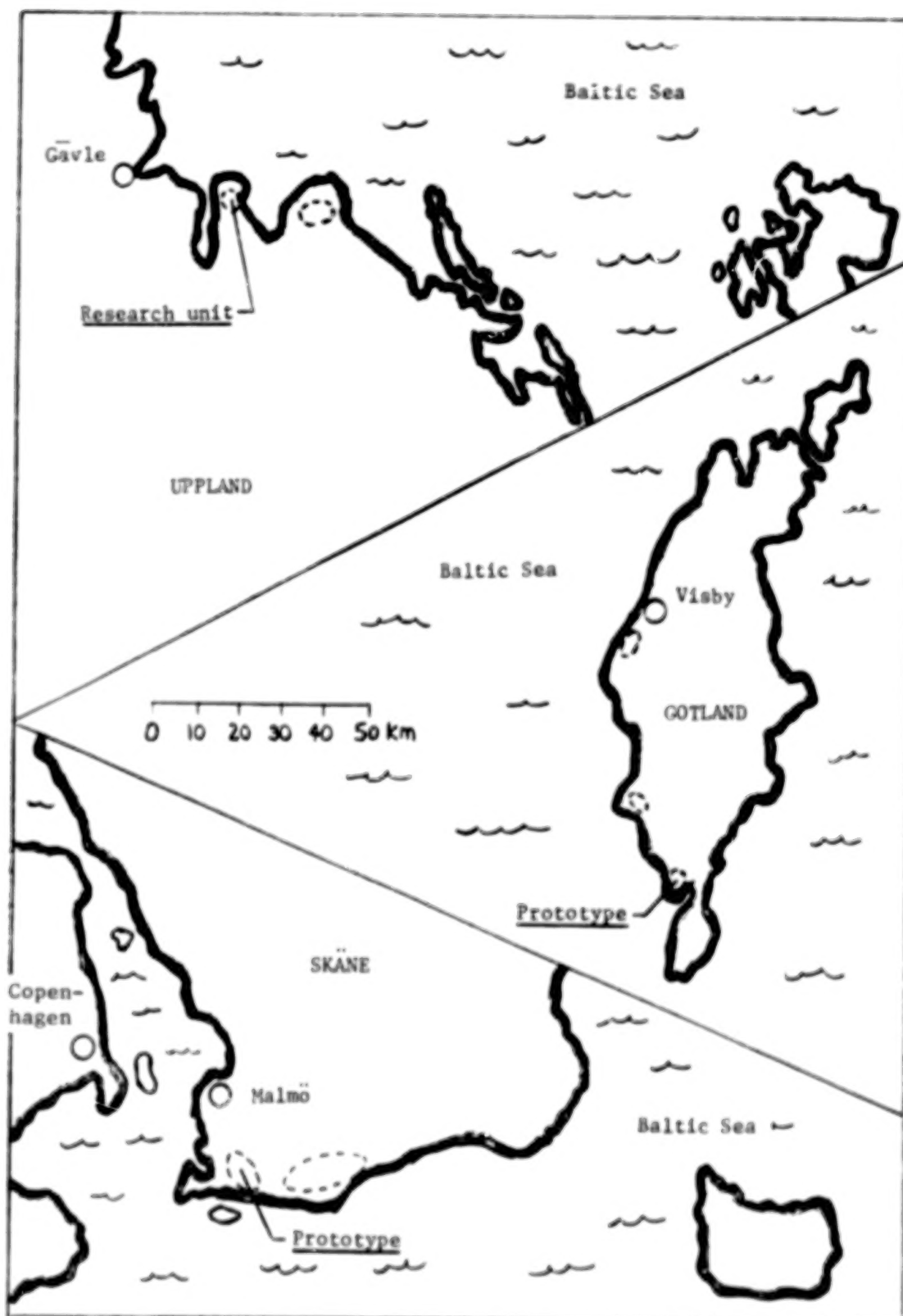


Figure 3. - Prototype sites.



## THE DANISH LARGE WIND TURBINE PROGRAM

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### SUMMARY

A short account of the Danish wind energy program and its present status is given. Results and experiences from tests on the Gedser windmill (200 kW) are presented. The key results are presented from the preliminary design study and detailed design of two new WECS (630 kW each). These two new WECS are planned to go into operation in mid-1979. The Tvind project (2 MW) is briefly mentioned.

### INTRODUCTION

The Danish Wind Energy Program was initiated 1977 and covers a four year period, up to January 1981. The total budget amounts to 41.4 Md.kr. or approximately \$8M. Of this amount, 36.4 Md.kr. is used for development of large turbines. The different tasks of the program and their costs are delineated in table I.

The objective of the Danish program can be briefly stated as:

- Get fundamental answers on the feasibility of wind power in a utility grid. Hence, use demonstration units, not merely a test facility.
- Get better ideas of costs. That means building the demonstration units large enough so that reliable extrapolations to large sizes can be made.
- Get better ideas of reliability, maintenance costs and expected lifetime. Therefore, use two units of approximately similar design.
- Get answers fairly quickly.

### TESTS OF THE GEDSER WINDMILL

The Gedser windmill, designed and built in 1956-57, was refurbished and operated from November 1977 to April 1979. The main characteristics of the windmill are stated in table II, and the general appearance can be seen from figure 1. Meteorological data were obtained from an instrumented mast situated 25 m to the west of the windmill.

The tests performed were intended to:

- Give information on the dynamic behavior of a stiff, three-bladed rotor
- Determine the power curve for this stall-regulated machine.
- Give information on the power quality obtainable with an induction generator.
- Gain experience in measuring techniques.

Since the tests have just ended, it is possible to give only some preliminary results. A full report of the tests will be ready in the fall of this year.

A sample power curve obtained from the 10 minute averages of wind speed and electric power output is shown in figure 2. Two-minute averages were used to produce the power curve on figure 3. In both figures, the calculated shaft power curves had no allowances for mechanical and electric efficiencies. Unfortunately, no high wind results were obtained. The maximum wind speed during test runs was approximately 18 m/sec. This means that the full effects of the stall-regulation have not been verified.

Figure 4 shows sample recordings of wind speed and electric power to indicate the magnitude of the fluctuations in electric output and also the influence of the averaging time used on the plotted results. A sample power spectrum is shown in figure 5.

The coupling to the grid during start-up presented no problems. Typically, the transient at the nearest transformer point would have an amplitude of 1.5 volts.

Valuable experiences on measuring equipment were gained. In particular, the transmission of data from the rotating parts to the ground station (by a telemetry system) turned out to be not as straightforward as was originally thought. The tests were interrupted on several occasions due to mechanical failure of different parts.

#### THE NIBE 630 KW DEMONSTRATION WIND TURBINES

The main effort of the Danish Wind Energy Program has been the design and construction of two 630 kW wind turbines. The main characteristics are listed in table 3, and drawings of the two, Model A and Model B, are shown in figure 6.

The main design features resulted from a four-month preliminary design study. A final design study was then carried out, and specifications were sent to manufacturers in February of last year.

Parts were ordered in May, and construction work is progressing according to plans.

The windmills will be operational in the latter half of 1979. It was decided at an early stage that the windmills should be quite similar. The main differences are that one (windmill A) is to be stall-regulated and provided with a stayed hub, while the other (windmill B) is fully pitch-regulated and with cantilevered blades. In most other respects the two windmills are similar.

The blades are of a mixed steel and fiberglass design. The outer 12 m of all blades are build up with a wound D-spar in fiberglass, around which is placed an outer shell, which also comprises the trailing edge. The shell is also fiberglass/polyester. Details of the rotor blades are shown in figure 7. The inner 8 m of a blade has a steel spar as the load carrying member and outer shells of fiberglass/polyester. Figure 8 shows the design of the junction between the outer and inner blade.

The choice of airfoil section (which should be the same on the two designs, so that the same mold could be used for all blades) was mainly determined to satisfy the needs of the stall-regulated machine. That is to say, an airfoil section was sought that gives a power curve as close to the ideal one as possible. The blade shape and twist of course also have an influence on the power curve.

Early in the study, the planform was chosen to be trapezoidal, and solidity, twist and airfoil section were the remaining parameters. The final result of the investigations was that NACA 44-series airfoil sections were preferable, and twist should vary linearly between tip and root with a total twist of  $11.7^\circ$ . Figure 9 shows the spanwise variations of chord, thickness ratio and twist.

The generator size was chosen to be around  $500 \text{ W/m}^2$  and an optimum tip speed of  $70 \text{ m/s}$  was found. For the stall-regulated machine it was found that four different pitch settings would be adequate. A starting position of  $15^\circ$ , a running position for wind speeds below  $10 \text{ m/s}$  of  $1^\circ$ , above  $10 \text{ m/s}$  of  $-4^\circ$ , and a brake position of  $-20^\circ$ .

Power and thrust curves for the final design are shown on figure 10 and 11. The calculated power coefficient versus inverse tip-speed ratio, i.e., wind speed over tip speed, is shown in figure 12. Constant power curves for different pitch settings and wind speeds are shown in figure 13. Also shown in figure 13 are the anticipated pitch-setting versus wind-speed variations for rotors A and B. Power duration curves are shown in figure 14. The annual output, not corrected for mechanical and electrical losses is calculated to be  $1,820,000 \text{ kWh}$  for windmill A and  $1,890,000 \text{ kWh}$  for windmill B. Expressed as mean power per  $\text{m}^2$ ,

the numbers are  $166 \text{ W/m}^2$  for windmill A, and  $172 \text{ W/m}^2$  for windmill B. These numbers are for a median wind speed at hub height of  $8.0 \text{ m/s}$ .

The natural frequencies for a Model B blade have been calculated and are shown in dependence of rotational speed on figure 15. Paragon Pacific Inc., under contract for the Danish program, performed a structural dynamic analysis of Model B using the MOSTAS code. A sample calculation result (fixed shaft MOSTAB analysis) is shown in figure 16.

The tower is designed as a reinforced concrete conical structure (fig. 17). The two lowest natural frequencies have been calculated as  $1.38 \text{ Hz}$  and  $7.34 \text{ Hz}$ , respectively. Measured in the scale of the rotational frequency of the rotor, the numbers are  $2.4 \text{ p}$  and  $12.8 \text{ p}$ . This means that during start-up, one passes through resonance with the  $3 \text{ p}$  excitation.

The nacelle layout, i.e., mainshaft, bearings, gearbox, brake, etc., is conventional as can be seen in figure 18. The yaw drive mechanism is hydraulically operated, and so is the pitch regulation system. Four yaw brakes are provided.

The costs of each of the two machines are very much the same. Table IV shows how the costs are distributed for the different parts. The estimated costs for the following units are not to be taken as the price for a series production machine, but only as an estimate for a limited production for a design which does not differ from the present one. Also it must be mentioned that no attempt was made in the present studies to find an optimum size wind turbine.

#### THE TVIND MACHINE

The Tvind windmill project is a private enterprise which was undertaken by a group of schools in Denmark. Technical specifications are listed in table V. The windmill went into operation on March 26, 1978, and trial runs are still being carried out.

The windmill is operating at variable rotor speed, and that part of the electric power which is fed into the utility grid ( $425 \text{ kW}$ ) has to pass through a rectifier-converter unit to get the frequency right. The surplus power is fed through a resistor bank and used for heating purposes. The heat reservoir with heating coils has not yet been installed, and accordingly the mill so far has only run partially loaded. The heating system is expected to be operational in April 1979.

The system and design as such, up till now has shown no major deficiencies. A lot of smaller defects, however, have had to be corrected during the trial runs. During initial runs, readings

were taken of the axial accelerations of the nacelle. Even when the rotor was passing through the speed where resonance with the first natural frequency of the tower (and the blades) was experienced, the measured accelerations did not exceed  $0.2 \text{ m/sec}^2$ .

One of the safety systems is blade-tip mounted parachutes, which are released when the centrifugal force exceeds a preset limit. The system has been tested and seemed to function satisfactorily. The automatic control system has not yet been operational, and all operations are done "by hand."

Noise emission, even when running at no load, is not negligible. In particular, the noise emitted when a blade passes the tower wake is quite noticeable.

#### DISCUSSION

Q. Has the Gedser machine been operated yet?

A. Yes, it has been operating but at a reduced power of 200 kilowatts of electricity.

Q. When do you expect to have the 630 kW machines in operation?

A. According to the plans, the first one will be up in the middle of June and the second one in September of this year.

Q. It appears in the photo of the Gedser wind turbine that the unit had acquired another set of cables since the earlier drawings of it. Is this an evolution aimed at restraining additional modes of bending of the blades?

A. Those cables were added ten years ago when it was running, and I am not quite sure why. What you are referring to might be the reason.

TABLE I. - DANISH WIND ENERGY PROGRAM, 1977 - 1981

	MdKr
Tests on Gedser windmill	3.3
Design and construction of two 630 kW WECS	20.4
Siting studies	1.4
Tests on two 630 kW WECS	6.0
Theoretical studies a.o.	3.6
Project management	<u>1.7</u>
Subtotal, large WECS	36.4 (\$7 M)
Small WECS development, test station	<u>5.0</u>
Grand total	41.4 (\$7.9M)

TABLE II. - MAIN CHARACTERISTICS OF THE GEDSER WINDMILL

Rotor location	Upwind
Rotor diameter	24 m
Number of blades	3
Blade tip velocity	38 m/s
Rotational velocity	30 rpm
Rotor area	450 m <sup>2</sup>
Blade construction	Steel main spar, wooden webs, aluminum skin. Heavily stayed. Braking flaps in blade tips
Regulation	Stall regulated, no pitch control
Generator	Asynchroneous 200 kW, 750 rpm
Transmission	Double chain 1:25
Tower	Stiffened concrete cylinder Hub height, 24 m
Performance	Self starting at 5 m/s 200 kW at 15 m/s Typical annual production, 350,000 kWh/yr
Weight of one blade	1650 kg



TABLE III. - TECHNICAL SPECIFICATIONS FOR MOD A AND B

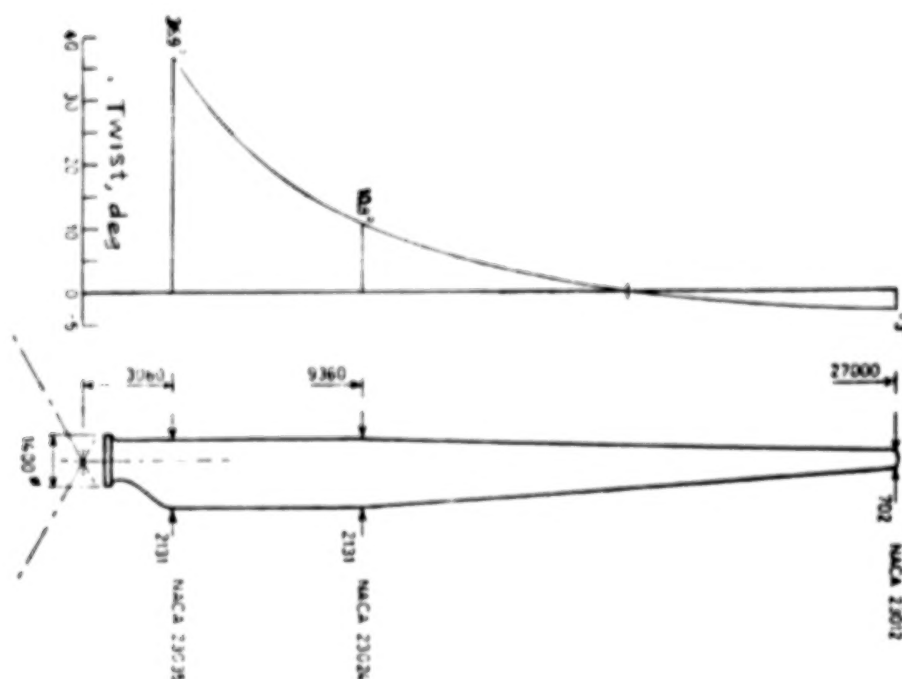
	MOD A	MOD B
Rotor diameter, meters	40	40
Hub height, meters	45	45
Tower height, meters	appr. 41	appr. 41
Rotor location	upwind	upwind
Number of blades	3	3
System life, yr.	25	25
Wind speed; cut-in, meters/sec	5	5
rated, -	appr. 13	13
cut-out, -	25	25
Weight of 1 blade, kg		3.370
Rotor speed, rad/sec	appr. 3.5	appr. 3.5
Rotor cone angle, deg	6	6
Rotor tilt angle, deg	6	6
Yaw rate, deg/sec	0.4	0.4
Pitch regulation:		
range, deg	+15 to -20	+90 to -1
maximum speed, deg/sec	6	8
normal speed, -	1	6
Generator: type	Asynch., 4-pole	Asynch., 4-pole
installed power	appr. 630 kVA	appr. 630 kVA
weight, kg	appr. 4000	appr. 4000
Transmission: type	conventional	conventional
ratio	appr. 1:45	appr. 1:45
weight, kg	appr. 10000	appr. 10000

TABLE IV. - PRICE BREAKDOWN, 630 KW WIND TURBINE

	Actual Price 1st Unit		Estimated Price Following Units	
	x 1000 d.Kr	%	x 1000 d.Kr	%
Acquisition of land and preparation of site	262	4	135	3
Electrical equipment and connection to mains	648	9	450	10
Tower	1,096	16	900	20
Nacelle, incl. rotor hub	2,822	40	2,000	44
Gearbox	260	4	250	5
Generator	135	2	125	3
Rotor, excl. hub	1,032	15	585	13
Assembly, running in	125	2	100	2
Consultants, rotor design	600	9	-	
	6,980		4,545	
	(\$ 1.330 M)		(\$ .866 M)	

TABLE V. - MAIN CHARACTERISTICS OF THE TVIND WINDMILL

Rotor location	Downwind
Rotor diameter and area	54 m and 2290 m <sup>2</sup>
Hub height	53 m
Number of blades	3
Max. rotational velocity	42 rpm
Blade construction	Fiberglass spar
Regulation	Pitch control
Generator	2 MW
Performance	Self-starting at appr. 5 m/s, 2 MW at appr. 14.8 m/s
Transmission	Conventional, ratio appr. 1:18.3
Tower	Reinforced concrete
Weight of one blade	5.2 kg
Weight of nacelle	100 kg





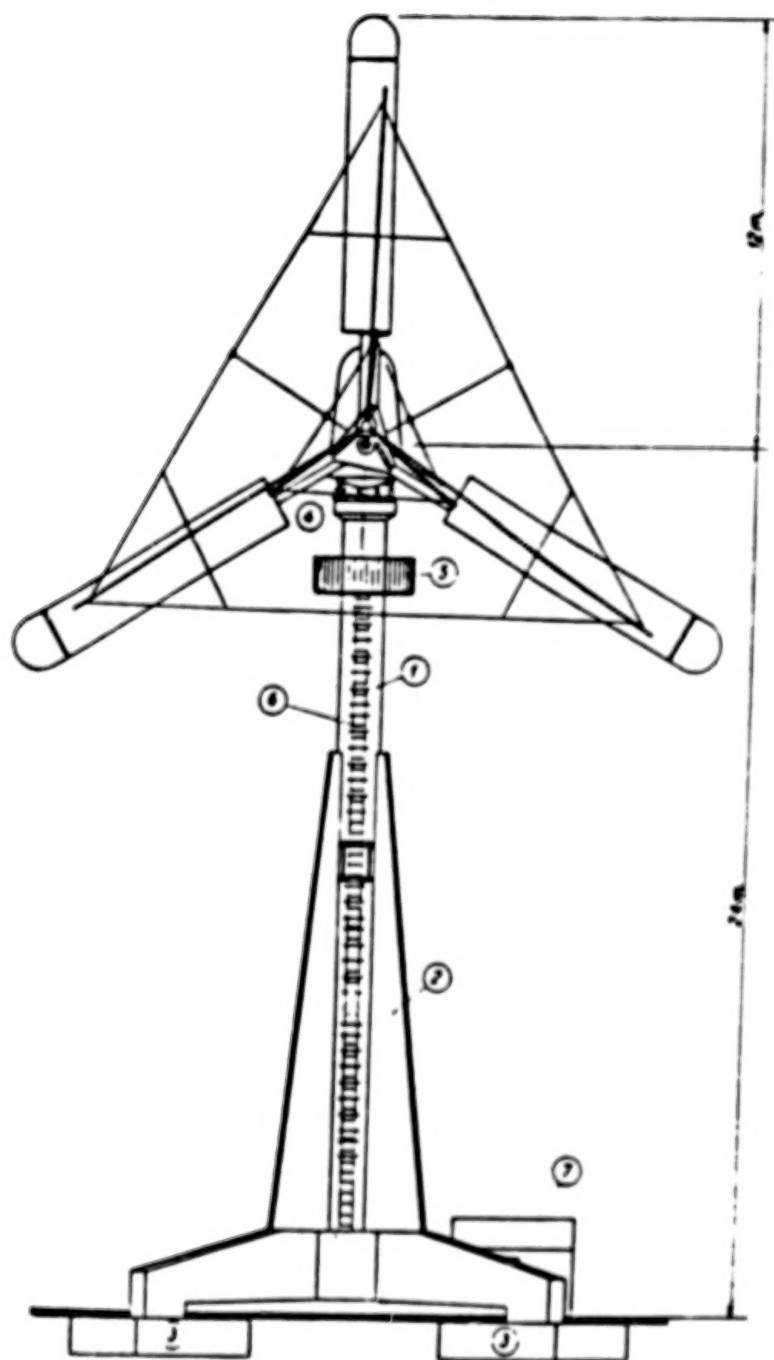


Figure 1. - The Gedser Windmill.

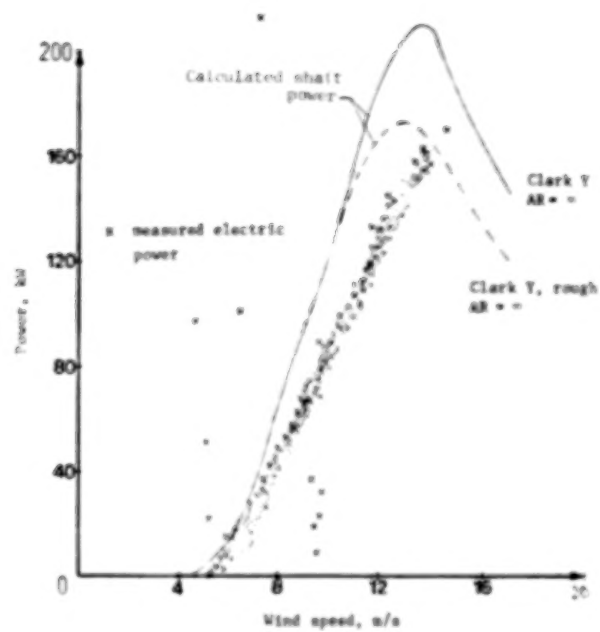


Figure 2. - A power curve for the Gedser Windmill. Block-averaged data for 10 minutes.

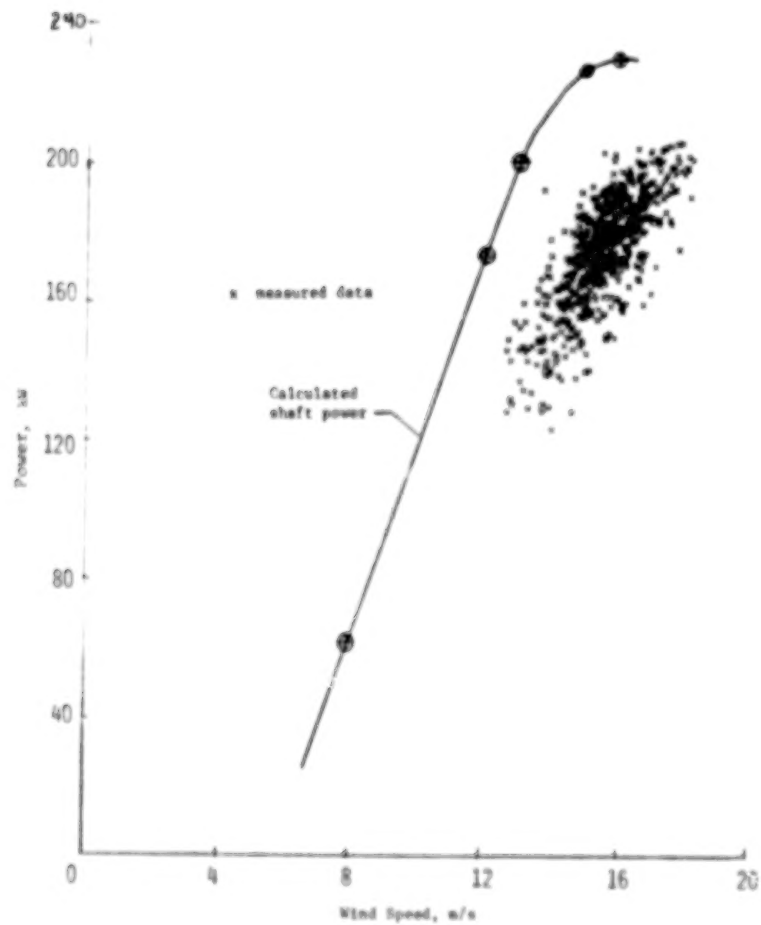
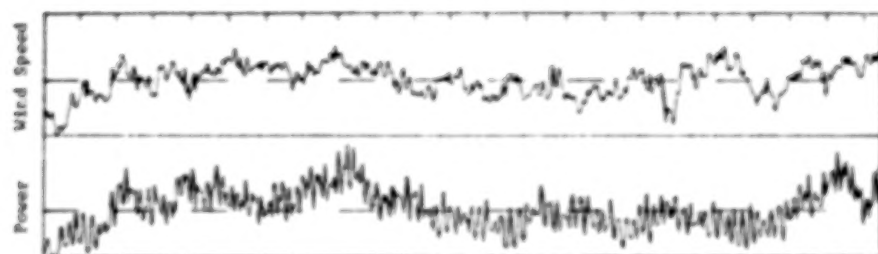
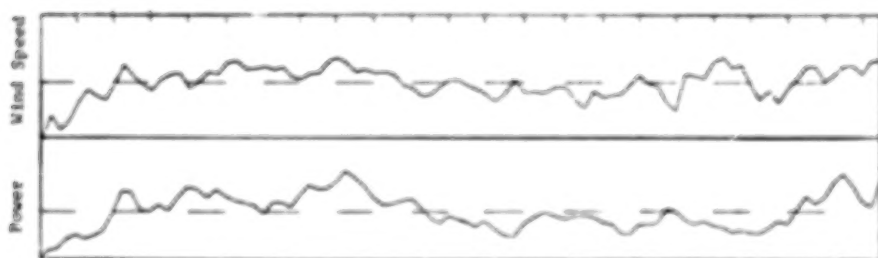


Figure 3. - Power curve for the Gedser Windmill. Data averaged for 2 minutes.



(a) Averaging time, 0.4 sec.



(b) Averaging time, 2.5 sec.

Figure 4. - Wind speed and electric power traces for Gedser Windmill showing power fluctuations and effects of averaging.

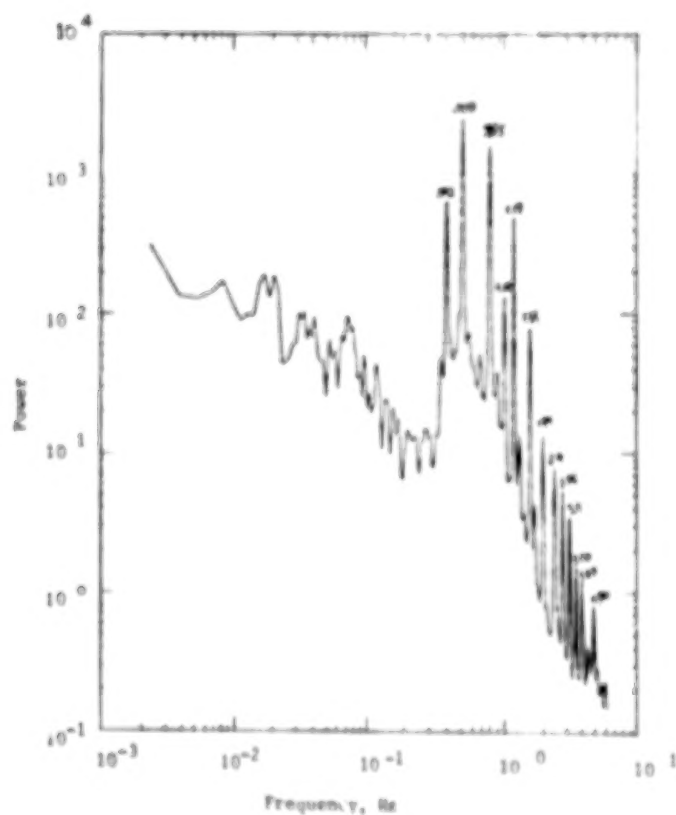


Figure 5. - Electric power for Gedser Windmill—power spectrum multiplied by frequency. Ordinate in arbitrary units.

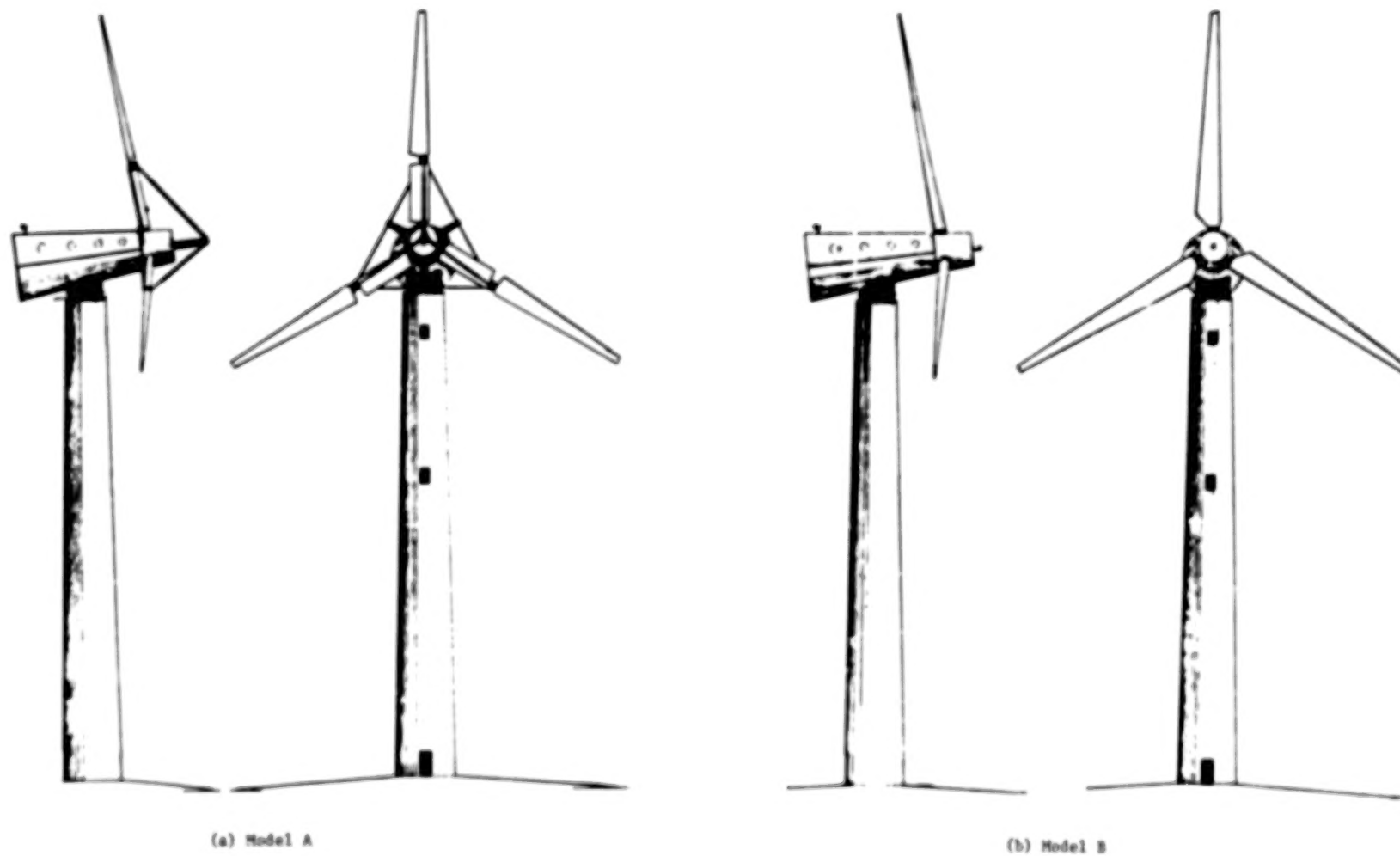


Figure 6. - Nibe 630-kW Demonstration Wind Turbines.

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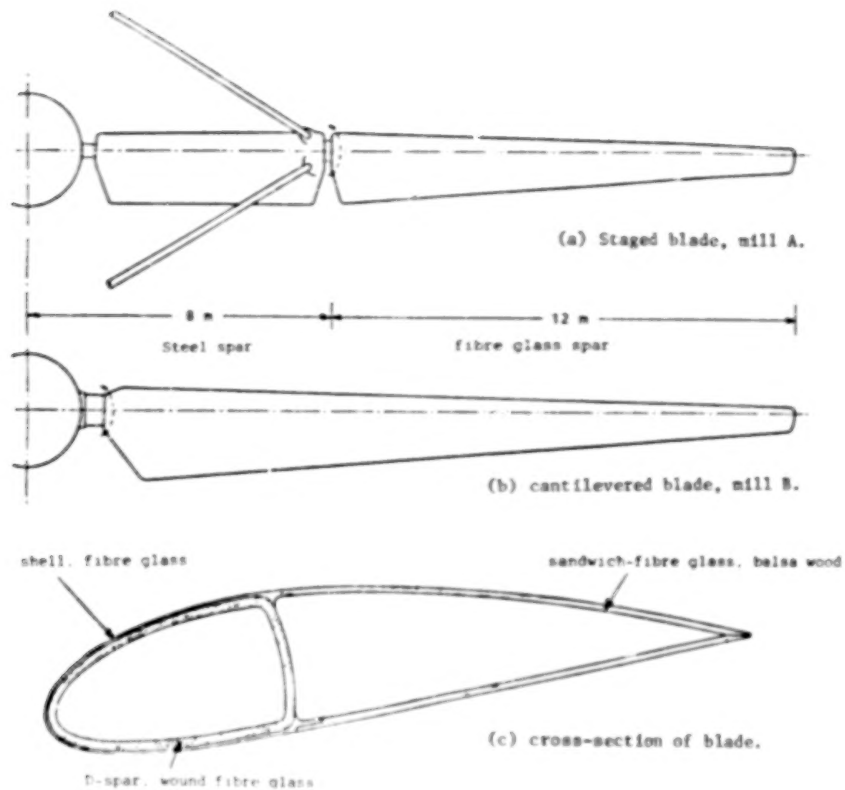


Figure 7. - Details of rotor blades. Nibe 630-kW Wind Turbines.

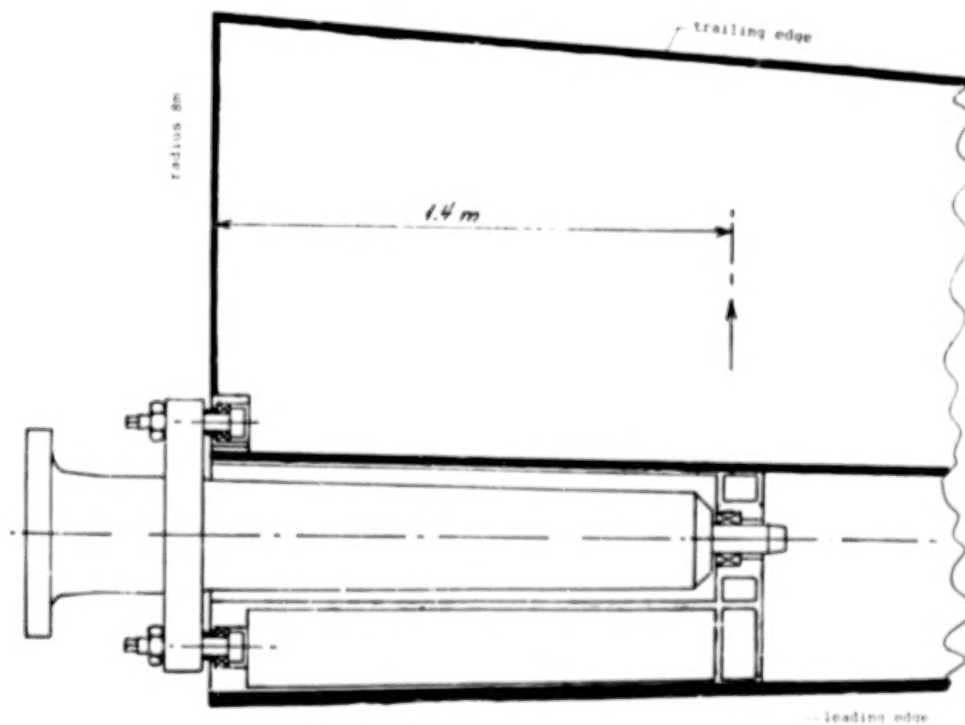


Figure 8. - Junction between outer and inner blade. Nibe 630-kW Wind Turbines.

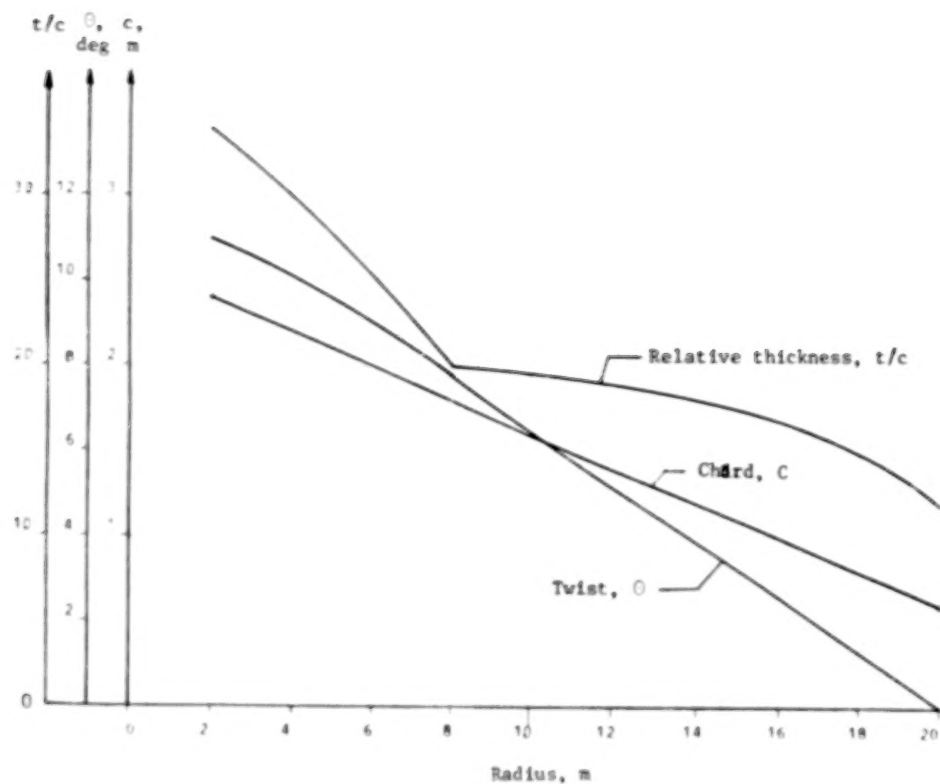


Figure 9. - Spanwise distributions for Rotor B, D = 40 m.

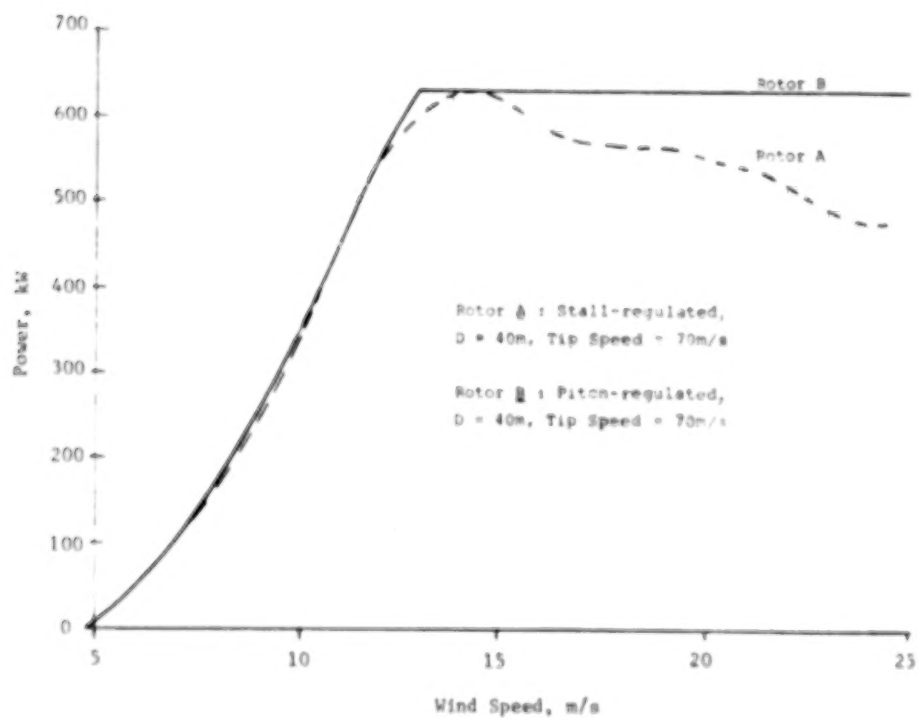


Figure 10. - Design power. Nibe 360-kW Wind Turbines.

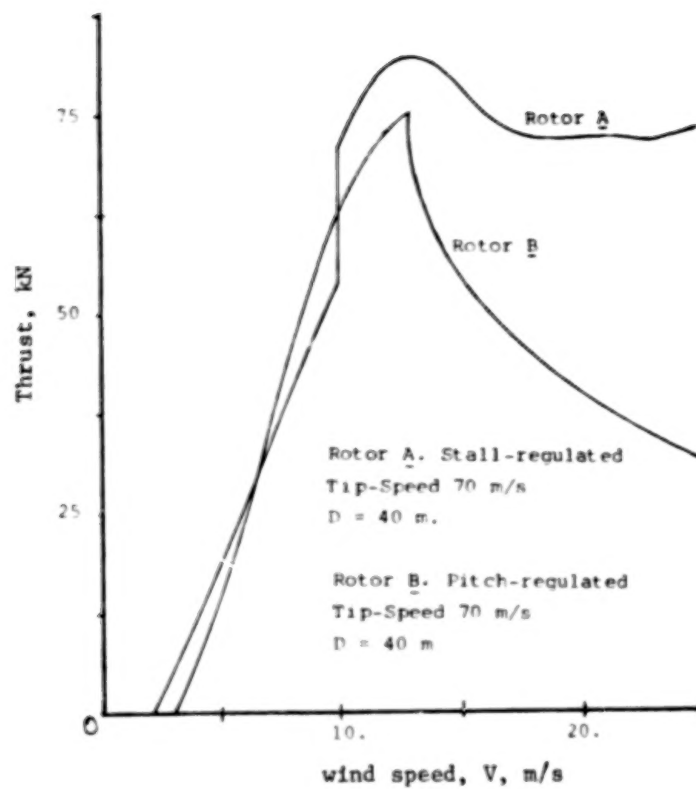


Figure 11. - Design thrust. Nibe 630-kW Wind Turbines.

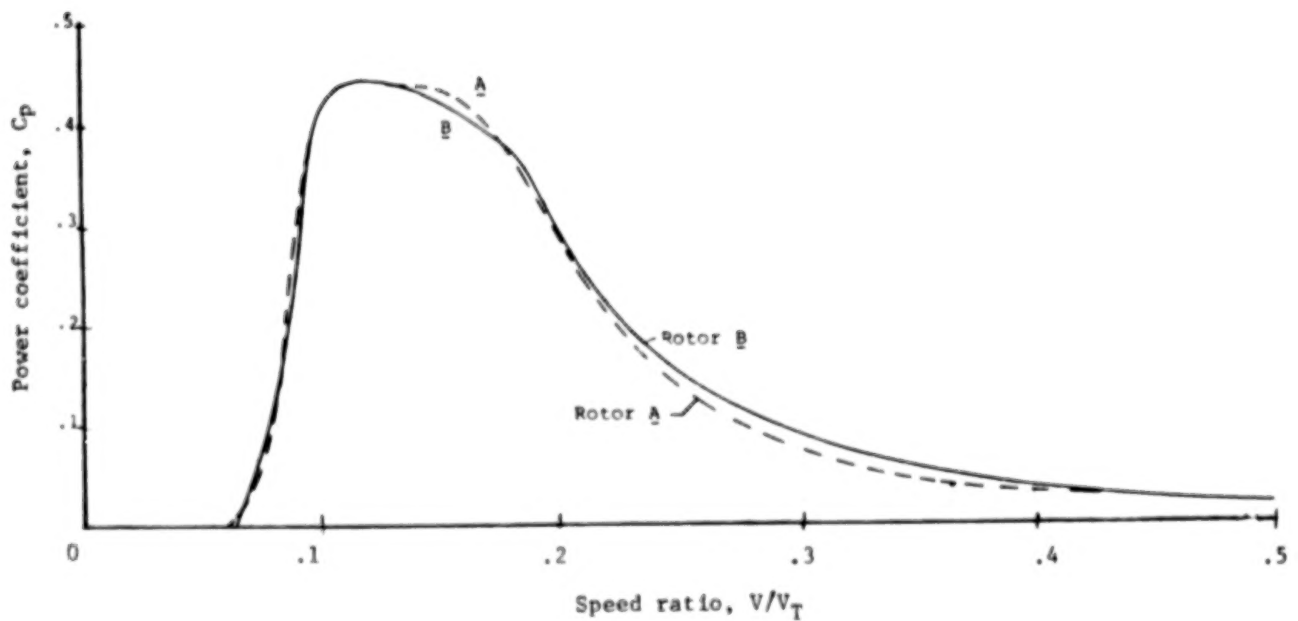


Figure 12. - Power coefficient curves. Nibe 630-kW Wind Turbines.



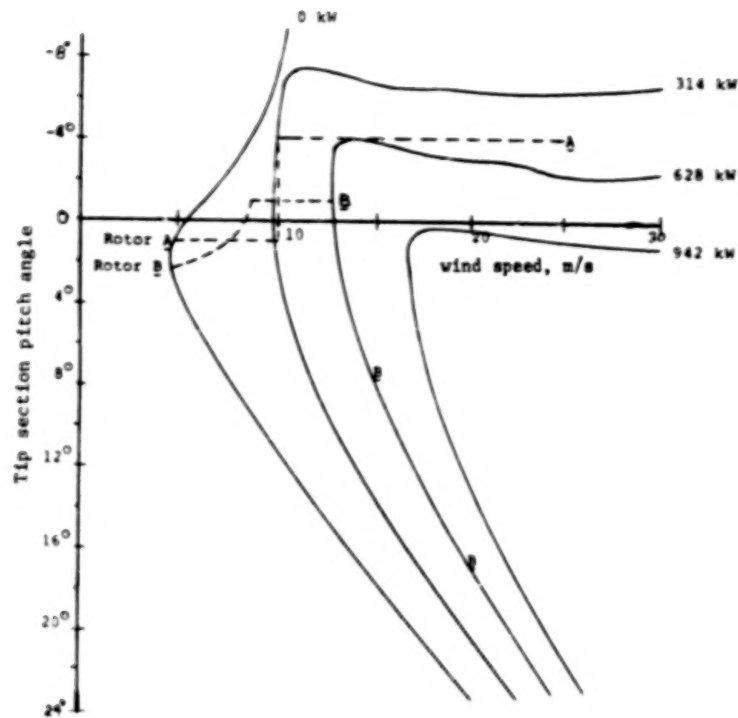


Figure 13. - Constant power curves with pitch regulation indicated.  
Nibe 630-kW Wind Turbines.

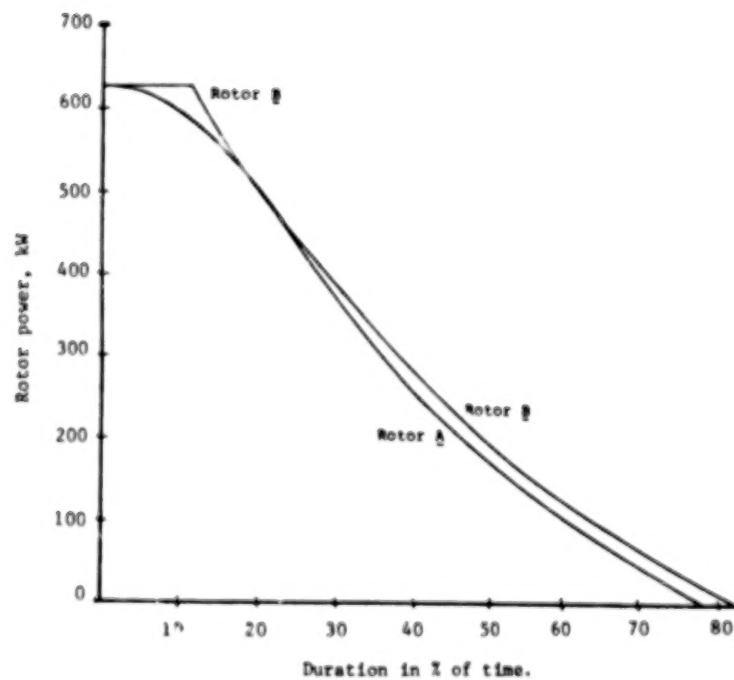


Figure 14. - Power duration curve. Nibe 630-kW Wind Turbines.

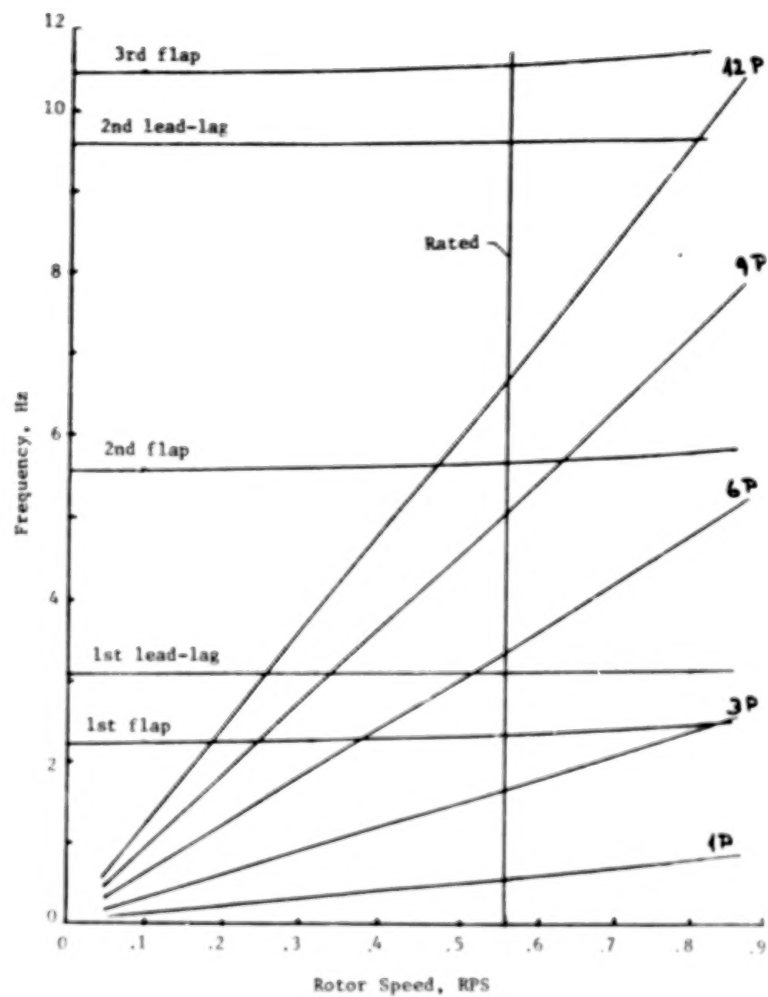
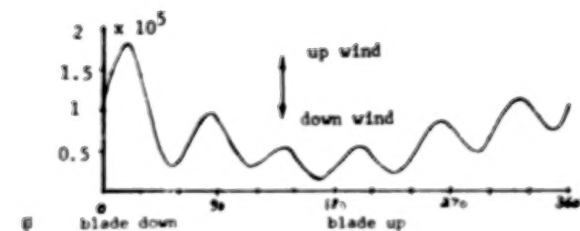
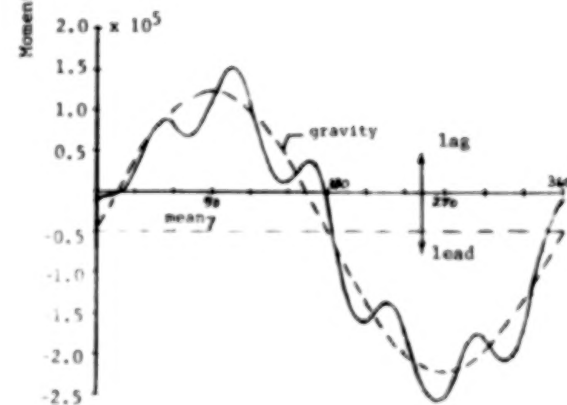


Figure 15. - Blade natural frequencies for Mod B blade.  
Blade mass, 3.365 kg.



(a) Flapwise bending



(b) In-plane bending

Figure 16. - Calculated blade root moments, Mod B. Conditions:  
 $\Omega = 3.5 \text{ s}^{-1}$ ;  $V = 22 \text{ m/s}$ ;  $\theta_p = 20^\circ$ ;  $\text{Yaw} = 10^\circ$ .

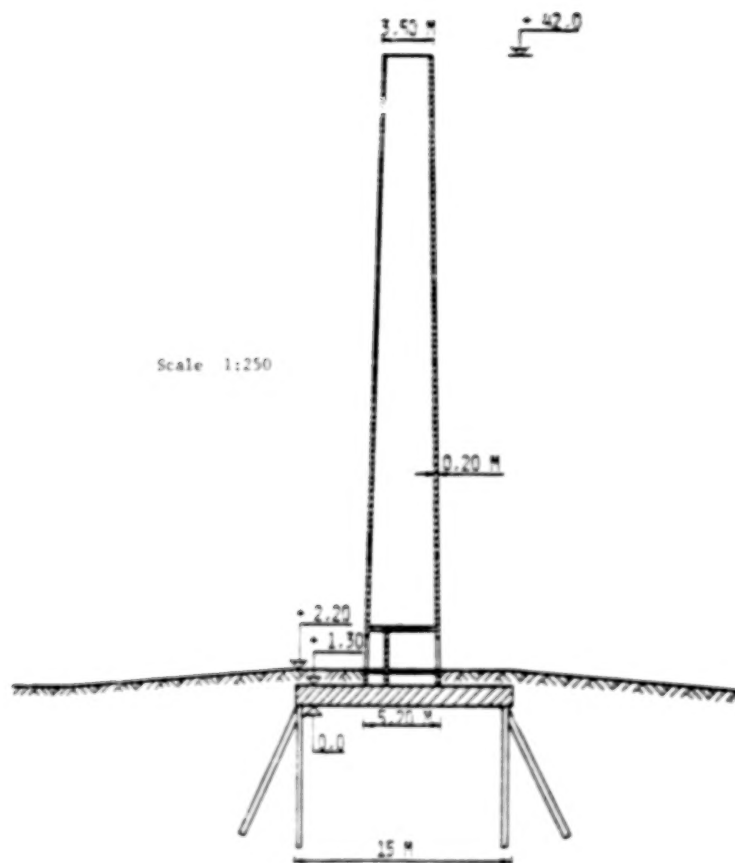


Figure 17. - Reinforced concrete tower. Nibes 360-kW Wind Turbines, Mod A and Mod B.

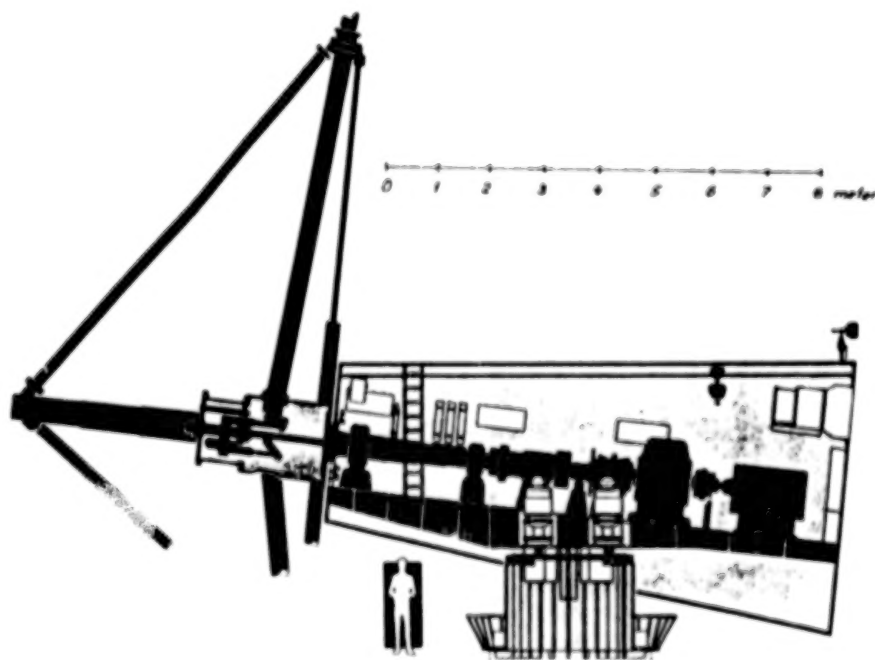


Figure 18. - Nacelle for Mod A. Nibe 630-kW Wind Turbine.

## LARGE WIND ENERGY CONVERTER - GROWIAN 3 MW

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### INTRODUCTION

The large wind energy converter GROWIAN has the function of producing electrical energy from the natural movements of the wind. A two-blade rotor mounted on the tower is rotated by the action of wind and transfers its power via a gearbox to a generator. The electrical energy thus obtained is then fed directly into the existing supply network.

The plant was designed with reference to a plant site in the North German coastal area. The design of the plant permits a long-term optimal exploitation of wind energy through the use of advanced, proven engineering techniques.

### MAIN FEATURES OF THE GROWIAN

The main features of the plant, as shown in figure 1, are:

- Two-blade rotor with pendulum hub
- Leeward mounting of the rotor
- Blade construction: steel-spar design with glassfiber airfoil
- Single stayed tower
- Controlled orientation of the nacelle and rotor into the wind

### SYSTEM DATA

Rated capacity	3 MW
Mean annual energy output	12 GWh
Power-to-area ratio	380 W/m <sup>2</sup>
Rated wind speed	11.8 m/s
Cut-in speed	6.3 m/s
Cut-out speed	24 m/s
Rotor diameter	100.4 m
Rotor speed	18.5 rpm $\pm$ 15%
Hub height above ground	100 m
Mass of tower head with rotor	240 t

## ENERGY YIELD

Calculations were based on an annual mean wind velocity of 6 m/s measured 10 m above ground. This average corresponds to the prevailing wind speed at a coastal site in the North German Plain. The installed capacity of 3 MW yields an annual energy output of 12 GWh. Figure 2 plots the power and the power duration curve. It can be seen that no power is produced 23% of the operating time owing to insufficient wind speeds. Forty-eight percent of the time the output falls below the rated capacity, while the plant operates at rated capacity 27% of the time. The plant is shut down 2% of the time as a result of excessive wind speeds.

The performance characteristics of the plant are shown in figure 3. The plant normally operates in the static control range. The extended limits of the dynamic control range are used for controlling brief fluctuations in the rotor speed. The rated capacity at nominal rotor speed is attained at a wind speed of 11.8 m/s. At higher wind speeds the surplus wind energy must be eliminated. To achieve this, the blade pitch is adjusted so that even at high wind speeds no more than the rated capacity is produced (fig. 4). The power output up to the cut-out point therefore corresponds to the output at the rated wind speed.

The efficiency of the aerodynamic conversion of a wind generator is characterized by the power coefficient ( $c_p - \lambda$ ) chart as given for the GROWIAN in figure 5.

## ENVIRONMENTAL IMPACT

The GROWIAN has no detrimental effects on the environment as it produces no noxious substances. The need for a pleasing appearance has always received special attention.

## CONSTRUCTION OF THE GROWIAN

As shown in figure 1, the wind converter comprises the main components of tower, nacelle, rotor, and their mechanical and electrical equipment.

### Tower and Nacelle

The tower is a slender cylindrical shaft of reinforced-concrete or alternatively of steel (fig. 6). It has an outside diameter of 3.5 m and a height of 96.6 m. The tower contains a spiral staircase, a lift, cable shafts, and the lifting cables for the tower head. Three pairs of cables are attached to the stay ring in the upper third portion of the tower and are strung to foundations in the ground.

The cylindrical nacelle is mounted atop the tower in such a way that it can be adjusted in yaw with reference to the wind. It accommodates the rotor, the gearbox, the generator, and various other units.

The shell-type welded housing has a diameter of 6 m and a total length of 22 m including the rotor mounted on the lee side. On the windward side a spar approximately 20 m long extends outwards, the tip of which contains the wind measuring instruments. Figure 7 shows a cross-sectional view of the nacelle. The collar extending downward houses the block and tackle system for raising the tower head, which is completely assembled on the ground. When raising the tower head, the tower shaft passes through the nacelle. The junction to the rotor is formed by the rotor bearing which is built into the hub support tube.

### Rotor

The GROWIAN employs a two-blade rotor 100.4 m in diameter. It is based on the principle of converting energy by utilizing aerodynamic lift. The requirements of strength and stiffness within practicable dimensions have been met for the first version by a steel-spar rotor blade 46 m in length which is assembled in three segments (fig. 8). The steel spar extends from the blade root to the blade tip and is given its airfoil form by mounted glass-fiber molded segments (fig. 9). The spar and skin of the blade were selected for their ability to withstand loads. The airfoil section was specially designed as a laminar airfoil. A rotor blade with a 35 m composite segment is presently being developed.

The blades are mounted on the pendulum hub and can rotate about their vertical axes on antifriction bearings so that their angle of attack can be adjusted by means of a motor-driven linkage in order to control the power output of the rotor. The pendulum hub protects components subjected to high loads such as the rotor blades, the tower, the tower bearing, and the nacelle from moments resulting from differential wind loads on the two blades. Excessive rotor speeds are precluded without resort to any extraneous power supply by means of a device actuated by centrifugal force which allows the rotor blades to turn to their feathered position.

### Mechanical Equipment

A planetary gear system with a ratio of 1:81 is used for stepping up the rotor speed of 18.5 rpm to the generator speed of 1500 rpm. The drive of the planetary gear is bolted to the rotor shaft. This system consists of two planetary stages and one spur gear stage and is bolted to the nacelle with a flange. A disk brake located at the high-speed output end of the gearbox is able to arrest the rotor revolving at low speeds or at full speed in an emergency, but only at the cost of its wearing parts. A universal shaft connects the gearbox output to the generator.

The tower head is turned into the wind by geared motors whose pinions engage in a gear ring integral with the tower.

## Electrical Equipment

The energy obtained from the wind converter is fed into the local supply network with due regard to economic considerations. Owing to the large fluctuations in the wind energy available, the most suitable generator is one which is not restricted to a set speed of rotation.

An asynchronous generator (3 MW), whose rotor is energized with alternating current by means of slip rings, lends itself well to this purpose, the frequency of the supply current corresponding to the difference to the synchronous frequency. This unit in the supply network behaves like a synchronous generator. By controlling the electric loading in the rotor circuit, any desired reactive and active current in the given range can be obtained.

A slipring system is provided for transferring the generated current as well as the monitoring and control signals from the rotating tower head to the electric cables in the tower. This system is located at the top of the tower concentric about the tower longitudinal axis.

## Servo Control

In order to make operation on a supply network possible, the fluctuations in the speed of rotation resulting from changing wind velocity must be reduced enough to allow the electric generator to process them. The control system has the function of starting the plant in compliance with a prescribed set of data, to keep it within the operating range, and to continue operation or shut the plant down, depending on the momentary wind flow and energy requirements. Furthermore, if the plant is operating in conjunction with the supply network, the frequency and the stator voltage must be held constant.

Power output and speed of rotation are regulated by controlling the blade pitch and the generator moment. The generator deviates elastically in the subsynchronous or hypersynchronous modes of operation whenever fluctuations in the rotor speed occur until the overriding speed regulator has reestablished synchronous operation. A schematic of the operating principles of the control system for power output and speed is shown in figure 10.

## Operational Control and Monitoring System

A programmable computer is provided for the operational control and monitoring of the converter. In addition to continuously checking the data, it is also responsible for registering and analyzing the operating condition of the plant and for detecting and signaling any malfunction.

## ASSEMBLY AND ERECTION

The steel tower as well as the reinforced-concrete tower are erected on a concrete foundation, which is laid at the site. The 9 m sections of the steel tower are set on top of one another with a crane, are bolted together, and then welded (fig. 11). The concrete of the reinforced-steel-concrete tower is formed with the aid of sliding molds. At the same time the plant building is erected.

When the tower has reached a height of 10 to 15 m, the shell components of the nacelle are placed over the tower shaft and are welded in place. The rotor shaft and the pendulum frame are positioned in their bearings with the aid of suitable lifting gear. The rotor is assembled in a horizontal position. The three blade segments are bolted and welded together in the steel sector. After the tower and the nacelle with rotor have been completed, the tower head is raised with the aid of the built-in hoist. The tower shaft passes through the nacelle.

## PLANT RELIABILITY

Special emphasis was placed on operational reliability in both the designing of the plant and the dimensioning of its components. The measures taken ensure that the CROWIAN is not endangered in its operation and does not constitute an environmental hazard even under the most inclement atmospheric conditions. Its stability complies with the specifications usually prescribed for high-rise structures. In the calculations of the static and dynamic loads arising from the action of the wind, gusts with a wind speed of up to approximately 60 m/s were taken into account.

## DISCUSSION

- Q. When will the CROWIAN be in operation?
- A. We expect to have it in operation in, roughly, two-and-a-half years.
- Q. Could you explain the emergency pitching apparatus in the event the electrical pitch control were to fail?
- A. As shown on that slide, there is a quick release bolt arrangement in the linkage connecting the pitch control actuator to the blade. In case of an emergency, it is activated and the blade becomes free. The blade then pitches itself to the feathered position by aerodynamic forces. The machine uses full span pitch control, as was specified by the government.
- Q. What is the weight of the machine and the blades?
- A. The weight of the machine is about 240 tons. The weight of the reinforced concrete is 750 tons, and each blade weighs 23 tons.



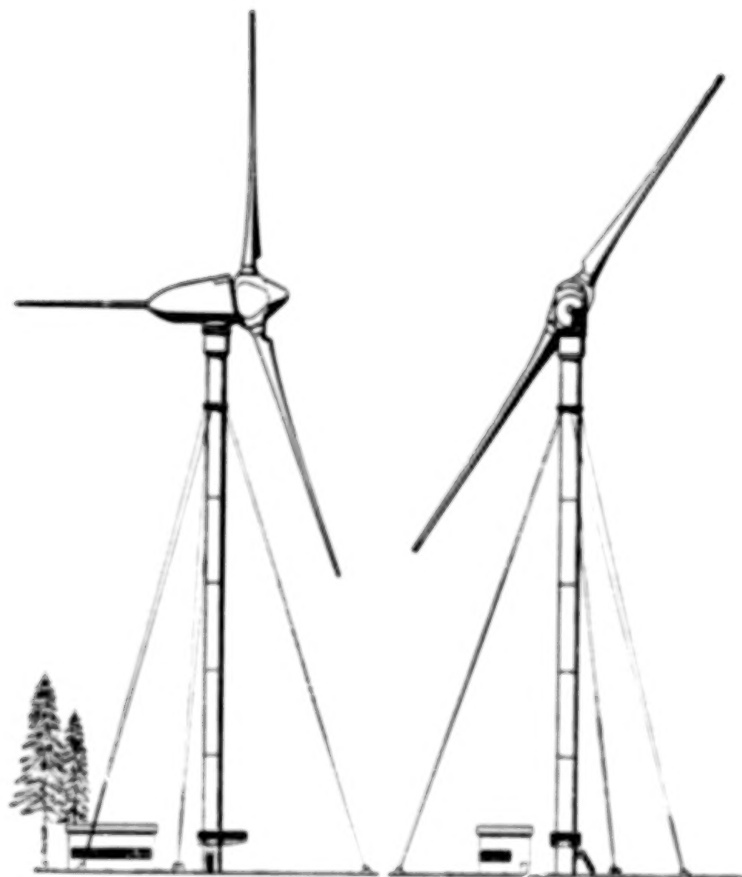


Figure 1. - Large wind energy converter GROWIAN 3 MW

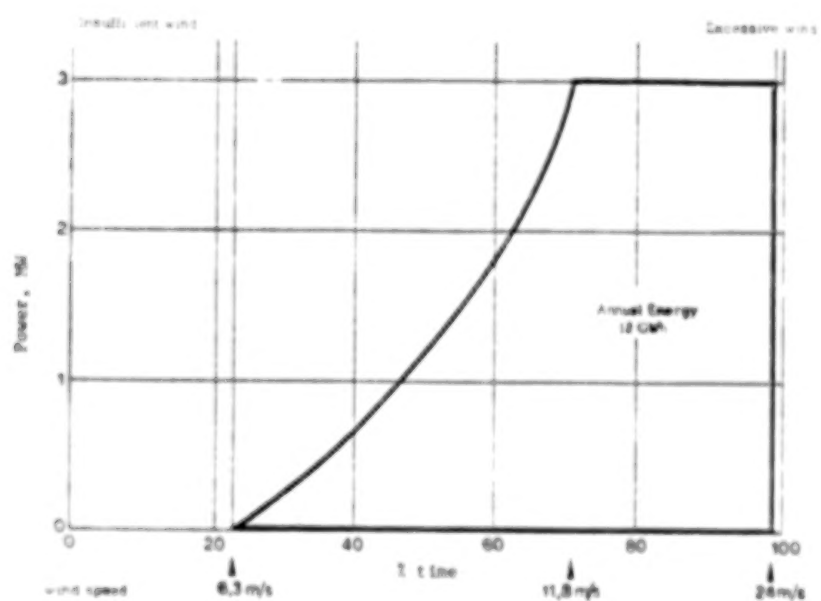


Figure 2. - Power duration curve.

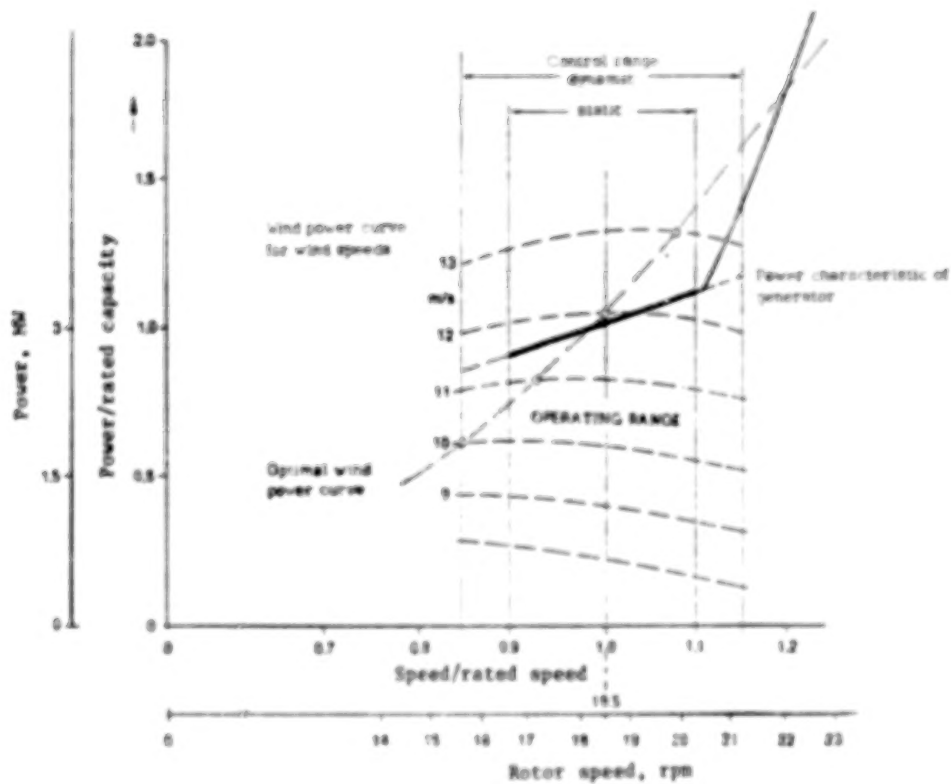


Figure 3. - Performance characteristics.

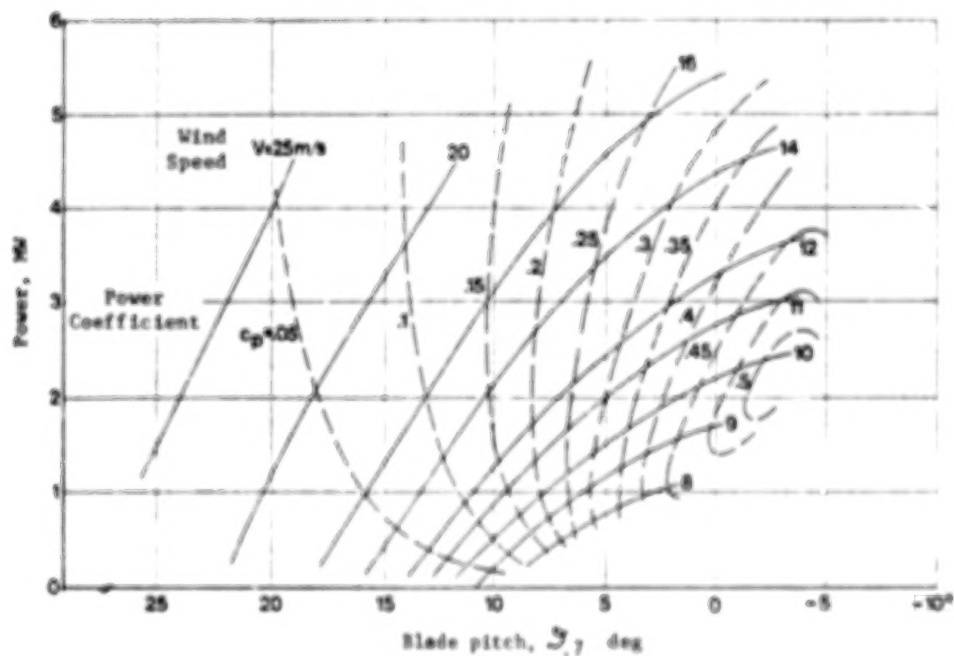


Figure 4. - Rotor performance.

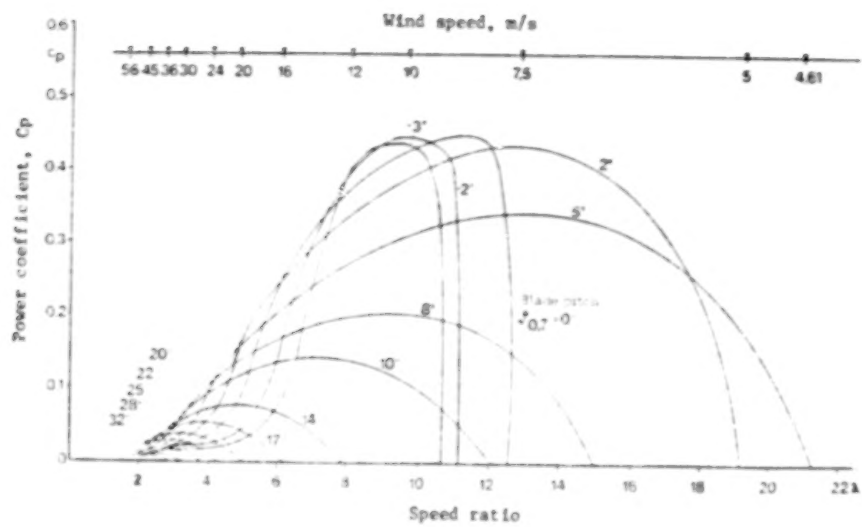


Figure 5. - Power coefficient chart.

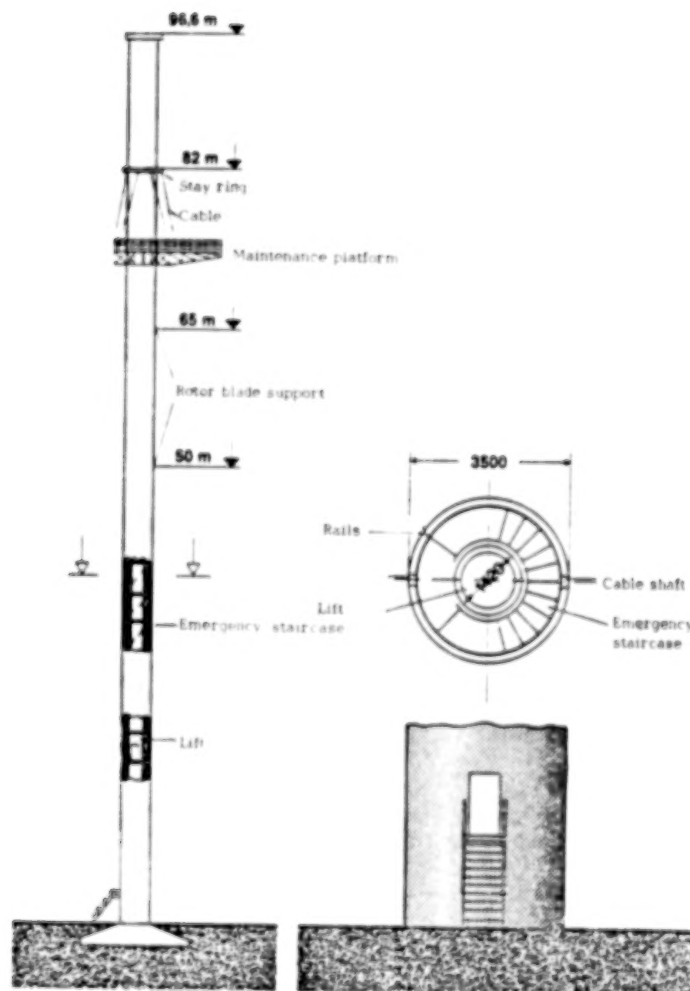


Figure 6. - Tower.

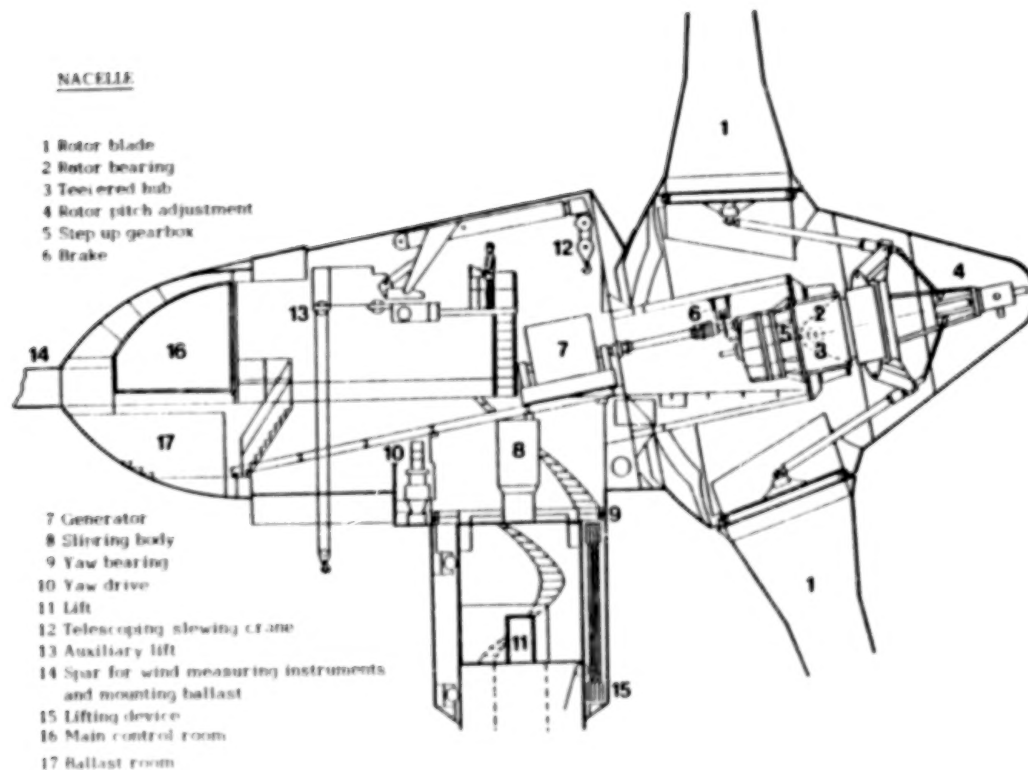


Figure 7. - Machine housing.

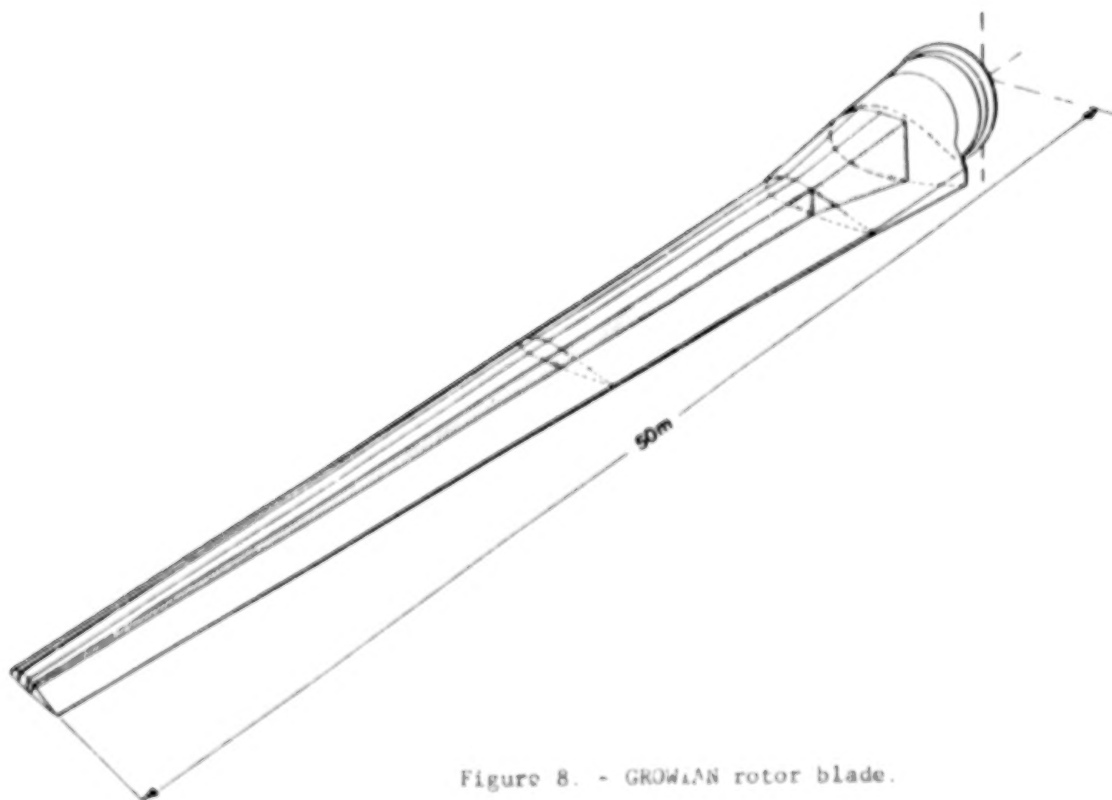


Figure 8. - GROWIAN rotor blade.

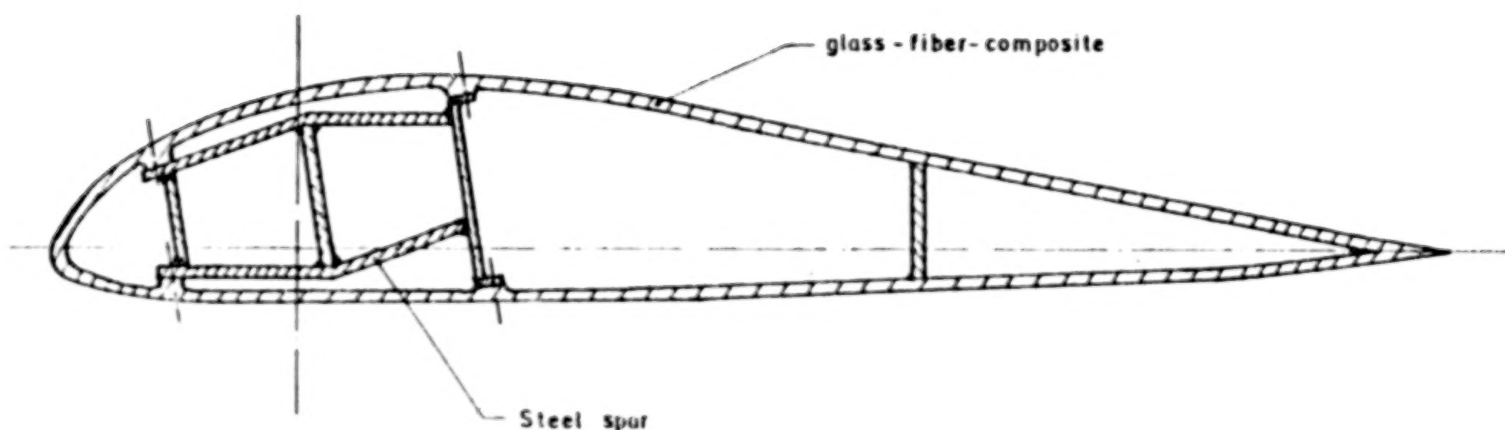


Figure 9. - Cross section of the rotor blade.

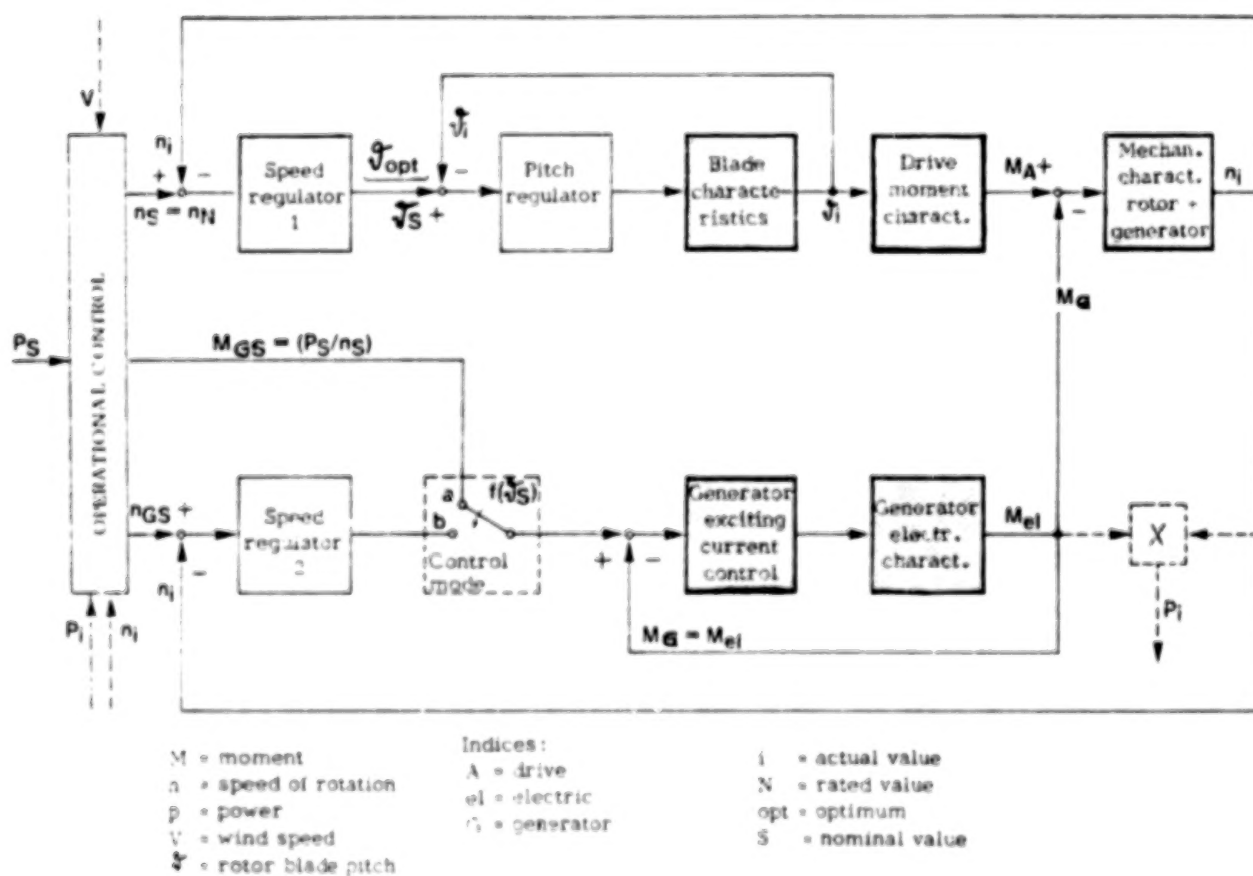


Figure 10. - Effective output-rotational speed control.

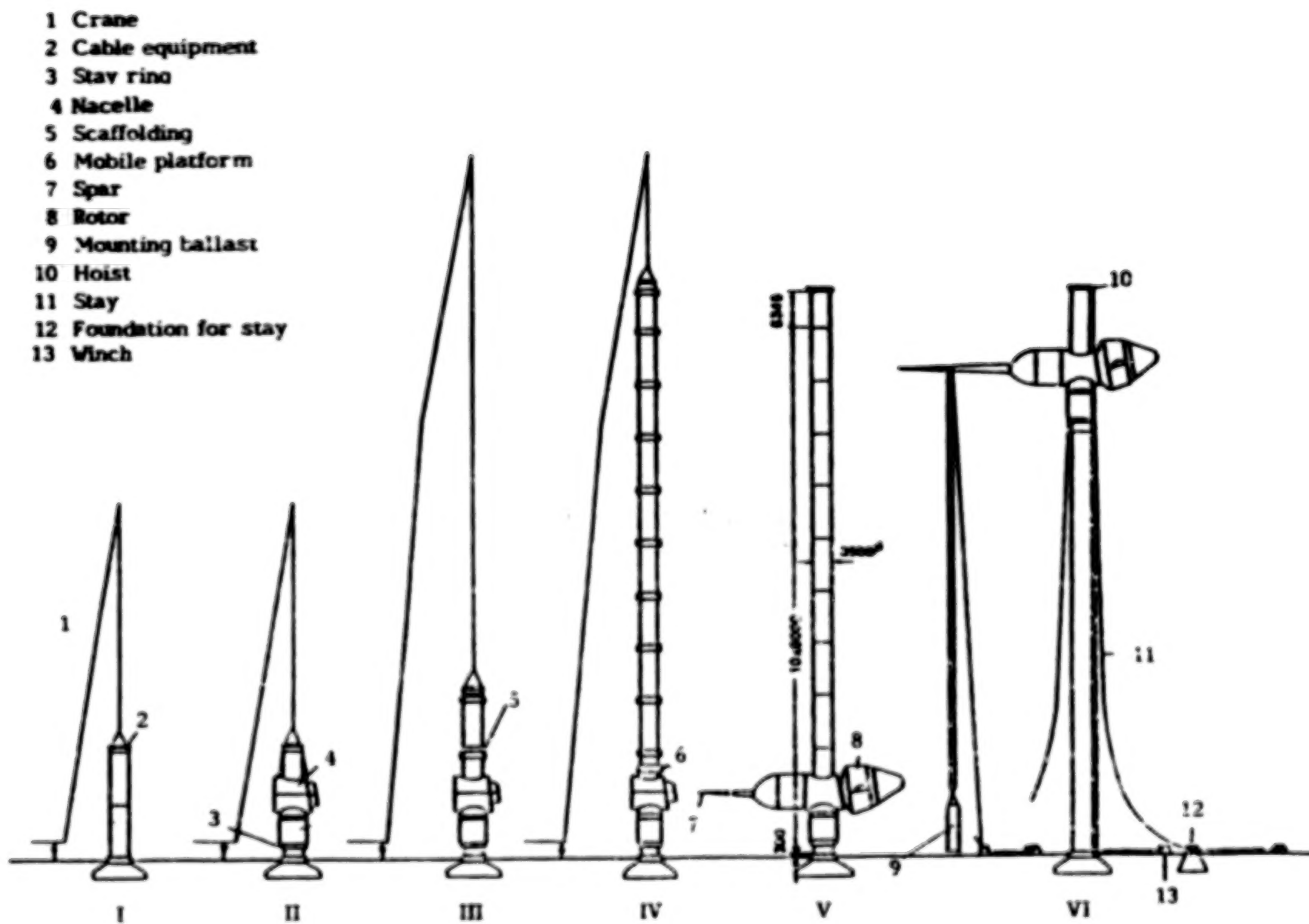


Figure 11. - Tower and machine housing assembly.

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## CHARACTERISTICS OF FUTURE VERTICAL AXIS WIND TURBINES (VAWTs)\*

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### Introduction

As a DOE facility, Sandia Laboratories is developing Darrieus VAWT technology whose ultimate objective is economically feasible, industry-produced, commercially marketed wind energy systems. The first full cycle of development is complete, and resulting current technology designs have been evaluated for cost-effectiveness<sup>1</sup>. First-level aerodynamic, structural, and system analyses capabilities have evolved during this cycle to support and evaluate the system designs. This report presents the characteristics of current technology designs and assesses their cost-effectiveness. Potential improvements identified in this first cycle are also presented along with their cost benefits.

### Current Design

#### Aerodynamics

The aerodynamic designs feature symmetric airfoils, starting with the NACA 0012 and now using the NACA 0015. The NACA 0018 has been used in some of the Canadian machines. Constant planforms are used over the entire length of the blade, and solidities (blade area/turbine swept areas) center in the 10 to 15% range for economic reasons. Recent test results promise 40% or higher maximum power coefficients.

Current designs use the inherent self-limiting feature because of aerodynamic stall ( $K_{pmax}$ ) at tip speed ratio of 3 or less. The corresponding maximum power coefficient ( $C_{pmax}$ ) occurs at a tip speed ratio of 5 to 6. Thus, regulation occurs when  $\left(\frac{Rv}{v}\right)_{K_{pmax}} / \left(\frac{Rv}{v}\right)_{C_{pmax}}$  or  $\frac{K}{m} = .5$  to  $.6$ .

These aerodynamic design characteristics yield turbines that are relatively efficient, can be manufactured by low-cost methods, and produce low-cost energy.

#### Structures

The structural characteristics of these designs are generally conservative. The blades have a uniform cross-section and end-to-end properties (Fig. 1). To account for uncertainties in the design and analyses, a margin of 2 is used between the calculated fatigue stresses and the allowable stress. These fatigue stresses are calculated for operation at 60 mph while the buckling response is calculated at 150 mph.

\*This work prepared for the U.S. Department of Energy, DOE, under contract DE-AC04-76DP00789.



Similarly, a factor of safety of 10 is used for tower buckling where conventional practice calls for a safety factor of 5. Current design philosophy is to set cable resonant frequencies above the possible excitation frequencies induced by turbine operation.

Current design towers are large-diameter, thin-wall steel tubes to minimize weight and cost. Fabrication tendencies have been to thicken the wall and reduce the diameter, making the towers more durable from a handling viewpoint. However, substantial weight and cost penalties are paid. The most cost-effective balance of weight, wall thickness, diameter, and ease of handling must be identified.

Blades are being designed using cross sections comprised of multiple extrusions (Fig. 1) except for blade chords of 24" or less, in which case a single extrusion is used. Multiple extrusions are joined by longitudinal welds whose chordwise location is chosen to minimize or prevent weakening of the blade cross section. These designs have used a constant wall thickness both chordwise and lengthwise.

The optimum rating of the current designs tends to be at a windspeed of approximately twice the annual mean, based on minimizing the cost of energy. These designs are two-bladed, have a height-to-diameter (H/D) ratio of 1.5, and a solidity of 12 to 14%. These designs yield about 10 to 12 kWhr/lb at a 15 mph mean windspeed and have a plant factor of approximately .25.

#### Cost Status

An economic analysis of this current design has recently been completed. The characteristics of the turbine were those previously stated; the turbines were considered to be in a grid application. A general configuration is shown in Fig. 2. Sandia Laboratories conducted this study, with A. T. Kearney, Inc. and Alcoa Laboratories furnishing actual cost estimates of several point designs. Alcoa and Kearney used these cost estimates to compute a profitable selling price for the individual point designs if they were manufactured, delivered, and installed by private industry.

Results are shown in Fig. 3.

These same results are plotted in Fig. 4 showing the effect of annual charge rate and dispatching.\*

Following are conclusions from this study:

- In production, the most favorable systems investigated apparently can provide utility electricity with a cost of from 4 to 6¢/kWhr with existing technology. Conditions associated with this estimate are a 100 Mw/yr production rate, 15 mph median windspeed,

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\*Dispatching refers to the standard utility procedure of regular inspection of machine output to record output, redirect output and check for abnormality.

90% machine availability, an 18% annual charge rate, a 0.17 wind shear exponent, and operation and maintenance (O&M) leveled with a factor of 2.

- The cost of energy decreases as VAWT rotor size increases up to the largest system investigated (1600 kW), largely because of the presence of costs that vary slowly or not at all with rotor size. Such costs are associated with O&M, automatic control hardware, and labor charges on all components. These slowly varying costs dominate the smaller systems and tend to limit their cost-effectiveness in this application.
- The cost of energy of all size systems is sensitive to the median annual windspeed and the annual charge rate for financing. Larger systems (above 100 kW) are sensitive to the wind shear exponent.
- The effect of production rate on the estimated selling price compares to a 90% learning curve.

Small systems in this application are less cost-effective. However, they do have certain inherent advantages over large systems. Among these are reduced development costs and technical risks, and lower capital investment requirements per unit. There are also markets that can use only small systems effectively: only because of energy demand limitations. These factors can increase the value-effectiveness of the energy produced by small systems. This potential should be recognized in assessing future significance of small VAWT systems as energy producers.

#### Future VAWT Design

##### Aerodynamics

Several aerodynamic changes are desirable to reduce the cost of energy. Several of these are to:

1. Increase maximum power coefficient.
2. Move the tip speed ratio associated with stall regulation ( $K_{pmax}$ ) closer to the tip speed ratio of the maximum power coefficient.
3. Increase the tip speed ratio of all points on the power coefficient curve.

These characteristics have been identified through the use of CPTAILR, an offshoot of the system optimization code VERS16.<sup>1</sup> CPTAILR can accept a six-parameter characterization<sup>2</sup> of a power coefficient curve for use in the optimization process. The cost of energy (COE) for changed aerodynamic characteristics was compared to that for standard characteristics.

Note that these preliminary investigations are being conducted to identify desirable features, estimate benefits, and establish goals and direction for future aerodynamic efforts. The low-cost 17 meter turbine was used as a test case for this investigation operating at sea level in a 15 mph median windspeed regime.

Changing the power coefficient curve to correspond to a change in  $C_{pmax}$  from .39 to .41 reduces the COE by 5%. The rated power is increased and the total energy increased while the operating speed remains unchanged. (Early test results using the extruded NACA 0015 blades on the 17 meter research turbine are showing maximum power coefficients of .41 to .42.)

Moving the stall or regulation tip speed ratio closer to the maximum  $C_p$  tip speed ratio increases the operating speed, drops the rated windspeed, and reduces energy costs by 8% for  $K/M = .7$ .

Shifting the power coefficient curve uniformly to a 25% higher tip speed ratio increases operating rpm and reduces energy costs by 2.5%.

The combined effect of increasing  $C_p$ , changing the regulation point, and shifting the  $C_p$  curve increases the rating, the total annual energy, and the operating speed, while reducing the rated windspeed and lowering energy costs by 14%.

These kinds of effects may be made possible by using cambered airfoils or nonuniform planforms on blades with little or no cost increases. Continued investigation of these potential changes will determine if inclusion in advanced VAWTs is feasible.

### Structures

Advanced structural requirements will be substantially reduced through the use of design requirements consistent with large horizontal machines, a more refined structural analysis capability, and the experience gained through a matured structural test program.

Probable changes in structural requirements will reduce:

- Parked buckling criterion for blades from 150 to 120 mph.
- Machine design/operational windspeed from 60 to 40 mph.
- Cable support system tiedown tension.
- Tower buckling safety factor, from 10 to 5.
- Blade weight, by tailoring blade wall thickness based on predicted operating stresses as a function of blade position.

These new criteria result in the following benefits:

Table I

<u>Item</u>	<u>Weight Reduction (%)</u>	<u>Cost Reduction (%)</u>
Blade Weight	50	~ 35
Spirally Welded Tubular Tower Weight	55	55
Generator/Electrical System		8
Transmission		10
Foundation and Tiedown		45
Shipping and Assembly		30
Total Net Reduction in Cost of Energy		25

Blade weight is reduced by approximately 50% and blade cost by 35% based on the use of several aluminum extrusions welded longitudinally. These extrusions would have wall thickness tailored for chordwise location. See Fig. 5. The weight reduction should also apply blades fabricated using steel or composites.

The weight and corresponding cost of the spirally welded tubular tower are reduced by 55%.

Because of the change in the relative costs of components, the light systems optimize at lower rated power and windspeed. This results in reduced generator/electrical and transmission costs. The generator/electrical costs are reduced by 8% and the transmission cost by 10%.

Since the total system weight and cable tension are reduced, the foundation and tiedown costs are reduced. This cost reduction is estimated to be 45%. Accordingly, the shipping and assembly costs reduction is estimated to be 30%.

The net effect of the new structural requirements is to reduce energy costs by 25%.

#### Transmission Investigations

In the existing technology designs, the transmission or speed increaser represents 15 to 20% of the total installed system costs. Reduction of the structural requirements for the future VAWTs changes the balance of costs so that the transmission's share of the total cost is 25%. Since the transmission costs are for standard hardware applied in a conventional manner to wind turbines, a new look at the speed increaser design and the application rationale is warranted. Topics such as design requirements, service factors, torque ripple, and cumulative damage will be examined in an attempt to better match speed increaser capability with wind turbine system requirements.

#### Improved Blade Fabrication

While the cost of blades fabricated from aluminum extrusions is expected to be \$3 to 4/lb, improvement in these costs would enhance the like-

likelihood of success of wind energy conversion systems. Candidates include improvements in the joining/extrusion methods and the use of other materials such as composites or steel.

Since the VAWT is amenable to the use of a constant planform, the pultrusion process for a glass/resin composite may be suitable for fabrication of VAWT blades. This process has been suggested in the past and may be a candidate for low-cost investigation.

Roll/stretch formed steel has also been suggested as a low-cost blade fabrication method. See Fig. 6. This process is also suitable for fabricating constant planform blades and uses a cheap, abundant raw material.

#### Summary of Cost Status

Better aerodynamics (.41 maximum power coefficient and moving the stall tip speed ratio to .7 of the tip speed ratio at  $C_{pmax}$ ) and future structural requirements combine to produce the following economies:

Solidity	12-14% No Change
Operating Speed	Increased by 30%
Rated Power	Reduced by 20%
Annual Energy	No Change
kWhr/lb System Wt.	20
Plant Factor	.30
Cost of Energy	Reduced by 35-40%, 2.5 - 4.0 Cents/kWhr for 15 mph, 18% ACR, O&M Factor 2.0

#### Conclusion

The existing technology for VAWT yields energy costs which are of interest. Improved technology (second generation) VAWTs show promise to achieve competitive energy costs through the use of improved aerodynamic and structural techniques.

#### References

1. SAND78-0962, Economic Analysis of Darrieus Vertical Axis Wind Turbine Systems for the Generation of Utility Grid Electrical Power, Volume I, Executive Summary; Volume II, Economic Optimization Model; Volume III, Point Designs; and Volume IV, Summary and Analysis of the A. T. Kearney and Alcoa Laboratories Point Design Economic Studies, to be published.
2. P. C. Klimas and R. E. Sheldahl, Four Aerodynamic Prediction Schemes for Vertical Axis Wind Turbines: A Compendium, Sandia Laboratories Report, SAND78-0014, June 1978.

# Discussion

- Q. I would like to know what is the investment cost at present and after production. Also, what is the installation cost of that machine, and how long would it take to install it?
- A. I am not prepared to answer all of those questions. The answers exist. As far as cost is concerned, roughly the installed cost is between 500 and 1000 dollars per rated kilowatt. For example, if I recall, the 1600-kilowatt machine cost is about \$700,000, while the 500-kilowatt design was around \$400,000. This is at the hundred megawatt per year production rate. The pre-production prototype cost is estimated to be about twice the continuous production cost. Mr. Ai will go over some of these numbers in his presentation.



Figure 1. Existing technology blade.

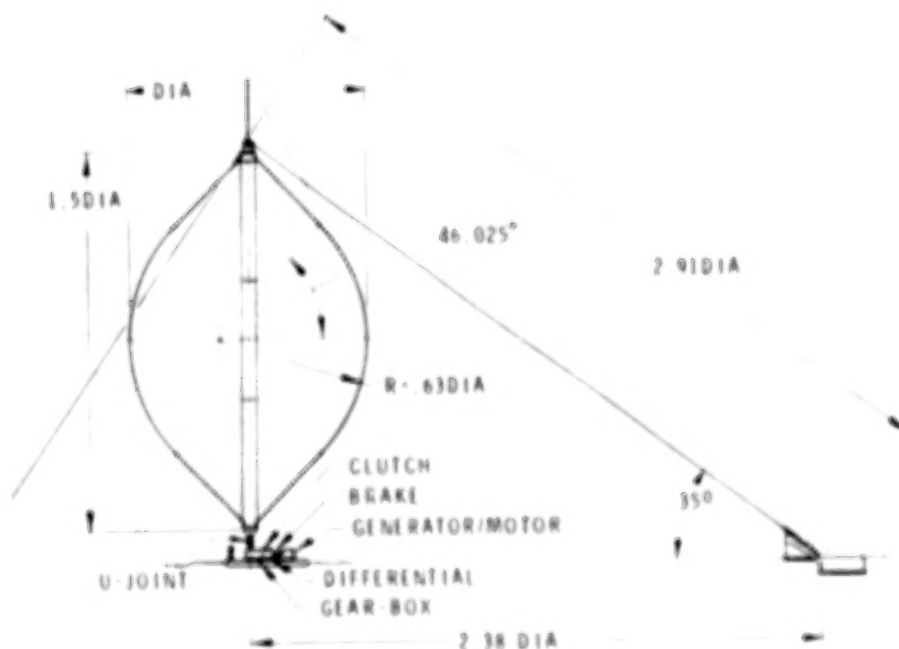


Figure 2. General configuration of turbine used in economic study.

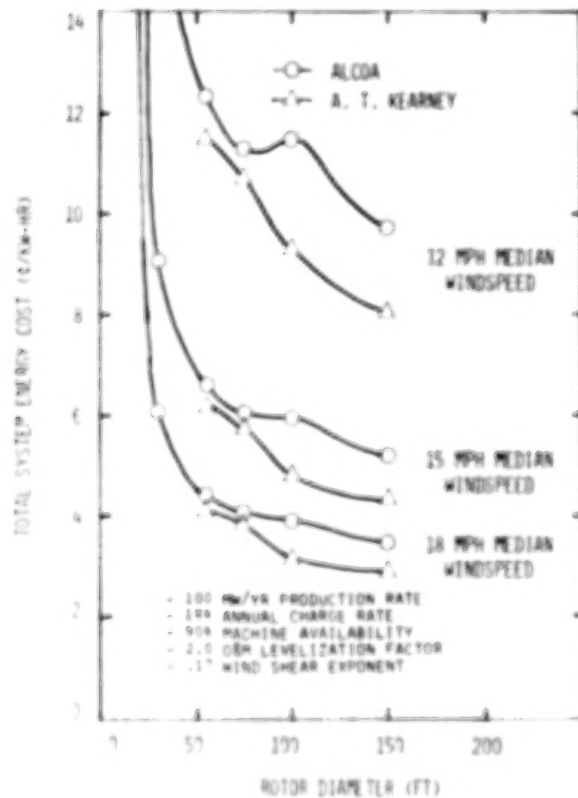


Figure 3. Total system energy cost for all point designs in three median windspeed.

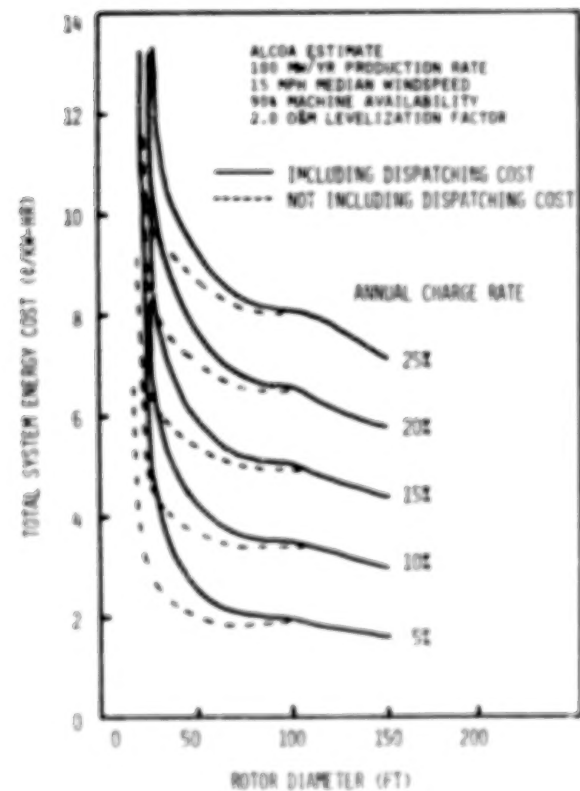
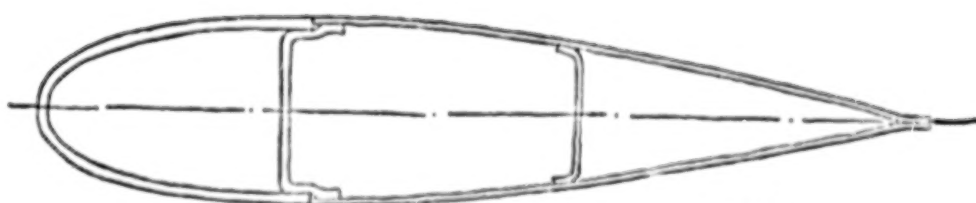


Figure 4. The effect of annual charge rate and dispatching costs on the cost of energy.





Figure 5. Variable wall blade section.



AIRFOIL SECTION

- ROLL FORMED STRAIGHT STEEL SECTIONS
- STRETCH FORMED INTO CIRCULAR ARC
- SEAM-WELDED STRUCTURE

Figure 6. Possible steel cross-section.



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## DESIGN CHARACTERISTICS OF THE 224 kW MAGDALEN ISLANDS VAWT

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### SUMMARY

The evolution of the main design features of the Magdalen Islands VAWT is described. The turbine has a rotor height of 120ft (36.58m) and diameter 80ft (24.38m). It was operated as a joint project between NRC and Hydro-Quebec in grid-coupled mode from July 1977 to July 1978 when the rotor was destroyed in an accident. The accident, although unfortunate, tested the basic integrity of the design in a gross overspeed condition, and the rotor is being rebuilt with minor modifications. Some directions for future VAWT research are suggested.

### INTRODUCTION

Early stages in the NRC laboratory development of the curved-blade Darrieus type VAWT have been described elsewhere (e.g. Ref. 1 and 2). The Magdalen Islands turbine (Fig. 1) is the largest machine of this type that has so far been built, and a recent paper (Ref. 3) describes some of the operating experience, including performance data, obtained during its field tests between July 1977 and July 1978. That paper also describes the accident that destroyed the rotor last July. The present paper is therefore limited to discussion of the factors that affected the choice of the main design features, and to some indications for future VAWT research and development effort.

During the accident, which occurred in moderate winds after the rotor had been de-coupled from the remainder of the drive train while the latter was undergoing maintenance, the rotor was subjected to several hours of runaway operation at rotational speeds up to 75 rpm, approximately twice the normal design speed. Failure eventually occurred in the base attachment of one of the four guy cables; no other structural failure occurred until the rotor struck the ground. The rotor is now being rebuilt without change in its basic structural design, for installation in early summer 1979.

Briefly, the early history of its development is as follows. By 1974, following field trials and wind tunnel tests of small-scale VAWT's, it had been decided to proceed with the early development of a large-scale, grid-coupled unit. Although some opinions had suggested that dynamic problems would limit large-scale VAWT development, simple dimensional analysis showed that most of these problems could be made invariant with scale.

It had also been confirmed in small-scale tests during 1973 that aerodynamic stalling of the blades in a constant-speed VAWT could be used to flatten or lower the power output curve beyond rated wind speeds, so that no variable geometry or associated automatic control was required. Although no request for proposals for a large machine had been issued, DAF-Indal Ltd, Toronto forwarded an unsolicited proposal to the Canadian government in December 1974, for a 200 kW turbine, based upon a 3-bladed rotor of height and diameter equal to 90ft (27.43m). That company had pioneered the use of hollow extruded aluminum blades in small-scale turbines, and proposed the same blade technology for the 200 kW VAWT.

Hydro-Quebec, one of Canada's largest power companies, had already carried out studies of the application of wind energy in certain parts of its system, and had decided to procure a large-scale commercial WECS for field trials on the Magdalen Islands. The Islands' 13,000 inhabitants are supplied with electric power by Hydro-Quebec's central diesel generating station which has a total installed capacity of 39 MW. Annual mean wind speeds in most of the islands and at the turbine site are consistently close to 19 mph (8.5 m/s). The collaboration of NRC and IREQ (Institut de Recherches de l'Hydro Québec) in the WECS demonstration project was thus a natural development. IREQ's contribution to the project has in fact been larger, in terms of manpower and cost, than NRC's, since they designed and built the turbine control system, foundations and all other site details, carried out turbine installation, and provided full-time site operating and maintenance crew.

#### PRELIMINARY DESIGN DRIVERS

The following factors mainly affected the initial choice of the basic turbine design parameters, and resulted in a number of changes to the original DAF-Indal proposal. From the outset it was felt that the time scale and costs of development should be reduced as much as possible, and therefore a high priority was given to overall simplicity of design. The principal turbine geometric parameters were initially chosen on the basis of very little analytic study in order to get detail design started as early as possible. Figure 2 shows a calendar of events, primarily during the design phase.

(a) Blade structure As already mentioned, DAF-Indal had proposed the use of extruded aluminum one-piece airfoils for the turbine blades. They had determined that the largest available extrusions would provide an airfoil chord of about 24 inches (0.61m), and it was probably this figure more than any other single factor that set the overall scale of the turbine. Minimum-weight extruded airfoils tend to have

approximately constant skin thickness and their mass centres are therefore located aft of the quarter-chord point. From the point of view of prevention of bending-torsion flutter in the wings of aeronautical vehicles, this would be an undesirable characteristic. In early 1975, the Sandia laboratories made available a film of a violent blade flutter that they had discovered during wind tunnel tests of a 3-bladed rotor. In this complex mode, blade plunge and torsion were coupled, and a subsequent approximate analysis indicated that the primary coupling between modes was probably due to coriolis forces, and that the airfoil chordwise mass centre location was relatively unimportant. However, a high blade torsional stiffness appeared to be desirable and as a result the airfoil section was thickened from the NACA 0015 used on previous small turbines to NACA 0018 for the Magdalen Islands rotor. The thicker section also increased blade bending stiffness and therefore the resistance to buckling in high winds with the rotor parked. The extruded airfoils have four integral spanwise spars, and weigh approximately 25 lbs per foot of length (37 kg/m). The blades, each about 150ft (45.7m) long, contain no internal transverse ribs, but consist of 4 lengthwise segments joined by steel bars bolted to the front and rear spars.

(b) Number of blades The number of blades was chosen to be two rather than three, in spite of the fact that this would lead to higher torque ripple and dynamic loads on the support structure. The reason was partly to reduce costs of construction, but mainly to simplify field erection: the rotor was assembled flat and lifted after assembly. Moreover, with two blades, but not with three, drive train elasticity produces an attenuation of the twice-per-rev lead-lag blade bending moment amplitude.

(c) Rotor height-diameter ratio All previous known small-scale Darrieus rotors had been built with rotor height equal to or less than the diameter. For blades of parabolic shape (a close approximation to the troposkein shape), it can be shown that unity height-diameter ratio leads to maximum total swept area for a given blade length. There are, however, advantages to increasing height-diameter ratio. For a given swept area and blade solidity, rotor rpm is increased, thus lowering torque and alternator-to-turbine speed ratio. The bending of blades to the troposkein shape is also simplified since the required curvature of the blades at the equator is approximately inversely proportional to the square of rotor height-diameter ratio. Another important factor was that, for an optimum guy cable slope of 40 - 45° to the horizontal, a height-diameter ratio of 1.5 permitted sufficient clearance between blades and cables without a long shaft extension above the upper blade attachments, thus reducing design bending moments in the central column. The final choice of  $H/D = 1.5$ , was not, however, based upon elaborate analysis of cost

optimization, and is not necessarily considered the best choice under all circumstances.

(d) Turbine swept area As already stated, the choice of the largest available aluminum aerofoil extrusions tended to set the overall turbine scale. However, swept area is proportional to the square of blade solidity for a given blade chord and number of blades. If the solidity is defined as the ratio of the total blade chord to the turbine equatorial radius, it is usually found from performance calculations and wind tunnel tests that the highest peak values of aerodynamic efficiency are achieved with a solidity of about 0.2. This value would have led to a turbine radius of 20ft (6.10m), a height of 60ft (18.29m) and a swept area of only 1,600ft<sup>2</sup> (148.6m<sup>2</sup>). The final choice of a swept area of 6,400ft<sup>2</sup> (594.6 m<sup>2</sup>) was felt to be necessary in order to demonstrate the feasibility of a turbine of significant size, although it was recognized that the resultant low blade solidity (0.10) represented a compromise in terms of the maximum achievable efficiency.

(e) Spoilers All of the small VAWT's that had been built in Canada prior to 1975 had been equipped with automatic, centrifugally actuated blade spoilers to prevent overspeed and it was decided to incorporate a version of these in the Magdalen Islands turbine. Since a mechanical disk brake was also installed, for use in all normal and emergency stops, the spoilers in this case were designed as a back-up brake to be used only in the event of mechanical brake or drive train failure. Wind tunnel test data (summarized in Ref. 2) shows that only small spoiler area (of the order of 1% of turbine swept area) is sufficient to destroy aerodynamic power at all blade-to-wind speed ratios, and they are thus an attractive device from this point of view. Their failure to prevent turbine overspeed in the Magdalen Islands accident (appendix to Ref. 3) was not due to size or to failure to open but resulted from unstable operation, which in turn was due to inadequate centrifugal mass unbalance.

However, as a warning to other VAWT designers, spoilers present several problems. Fail-safe mechanical design, which should incorporate some means of ensuring that all spoilers open together, is difficult. If they are expected to slow the rotor to blade speeds well below wind speed, they will be subjected to reverse flow during every revolution, which may lead to aerodynamic instability about their hinge line. Some means of automatic or manual re-closure must also be built in. In every turbine we have built or procured, we have debated the question of deleting spoilers in the interests of simplicity, and in every case have decided to retain them one more time, including the rebuilt Magdalen Islands rotor.

(f) Blade struts As shown in Fig. 1, the rotor blades are supported by two horizontal struts of double A-frame



configuration.

The design of the struts was evolved during the wind tunnel tests which are described below. They serve three main purposes: to stabilize the blades against compressive buckling in the parked, high-wind condition, to raise the blade critical flutter speed, and to provide a means of damping the possible "butterfly" mode of inter-blade resonance. In the butterfly mode, the blades oscillate out of phase in the lead-lag direction (one blade leads while the other lags), and the mode can be excited aerodynamically if its natural frequency coincides with an odd multiple of rotor rotational frequency. In constant-speed turbines this mode can be avoided by placing its natural frequency between, say, the 3P and 5P frequencies, or alternatively by providing some means of damping. In the Magdalen Islands turbine, damping was provided as insurance, although no butterfly resonance was detected during tests without damping. The method of providing damping was as follows. The horizontal struts are not rigidly attached to the central column, but to sliding rods that pass through bushings in the column. Thus, torque is transmitted to the column through the struts, but they are otherwise free to translate if butterfly mode oscillations develop. Hydraulic dampers are installed in parallel with the sliding rods inside the column. Other aspects of the strut development are described below in the context of the wind tunnel tests. It should be pointed out that it is by no means certain that some form of blade support struts are absolutely required on large-scale Darrieus type VAWT's, and they represent a fruitful area for future design simplification.

(g) Alternator type A commercial (Canadian General Electric) 300HP induction motor with a synchronous speed of 720 rpm was chosen for the Magdalen Islands turbine alternator. One of the main reasons for the choice of an induction rather than a synchronous alternator was that preliminary analysis carried out early in 1975 (item 2 in Fig. 2) had indicated favourable dynamic behaviour of the elastic rotor-drive train-alternator system. The induction alternator avoids the introduction of an additional mechanical stiffness or torsional resonant modes into the system, and in fact provides some damping of the shaft torque ripple. No problems with the alternator have been encountered during start-up or normal operation.

(h) Available methods of dynamic analysis There was one overriding design "driver", if that term is applicable, that had to be faced by the NRC laboratories at the time of the decision to proceed with the large turbine in early 1975. No computerized structural analysis methods were available in the laboratory that were applicable to the dynamic analysis of large-scale vertical axis turbines, nor were the resources available to develop them within a reasonably short time. On the other hand, NRC had available several low speed wind tunnels,

including the 30 x 30ft (9 x 9m) V/STOL tunnel. It was therefore decided to base the dynamic analysis upon a series of aeroelastic wind tunnel models. The tunnel program is briefly summarized below.

#### AEROELASTIC WIND TUNNEL TESTS

Three scales of aeroelastic models were built for wind tunnel tests. The first set of models, at about 1/50 scale, were non-rotating models with solid strap blades for the investigation of blade collapse in high winds with the rotor parked. Because gravity loads may contribute to compressive instability of the curved blades, these models and the tunnel wind speed were scaled so as to preserve full-scale values of dimensionless stiffness quantities and also the Froude number. The results, in dimensionless form are presented in Ref. 2. For the strutted blades of the Magdalen Islands wind turbine, the wind speed for blade collapse is estimated to be well in excess of the specified maximum design wind speed of 135 mph (60 m/sec). The model tests indicated that the addition of struts approximately doubled the wind speed for blade collapse with rotor parked at the most critical angle.

A 1/24 scale aeroelastic rotor model was built and tested in 1975 (item 5, Fig. 2), to measure rotor cyclic loads and their Fourier components. The same model was driven to flutter speeds with various blade configurations in order to determine the dimensionless blade and strut stiffness parameters required to avoid flutter. The results were reported in Ref. 2. These tests incidentally confirmed the insensitivity of flutter speeds to the chordwise location of the blade mass centre.

Finally a 1/4 scale aeroelastic model was tested in early 1976 (item 8, Fig. 2). In this, and also in the 1/24 scale model tests, Froude scaling was not preserved, since gravity forces have no cyclic components in vertical axis turbines. Instead, full-scale speeds, relative stiffnesses, reduced frequencies and stress levels were preserved. The blades and central column of the 1/4 scale model were strain-gauged at a large number of locations. Sample results, in dimensionless form, were presented in Ref. 2. Of particular relevance to the final design of the full-scale rotor was the discovery that the butterfly blade vibration mode could have a natural frequency close to an odd multiple of the full-scale normal operating rotational speed (38 rpm) but that damping was effective in attenuating blade stress levels. Accordingly, the strut design was modified to incorporate damping capability, as already explained. Cyclic stress levels measured on the full-scale turbine were found to be in reasonable agreement with those measured on the 1/4 scale model in the wind tunnel. A summary of the full scale dynamic stress data at the maximum-

stress location on the blades was presented in Ref. 4. The cyclic stress levels were found to be well within acceptable fatigue limits.

Although the NRC aerodynamics laboratory is wind tunnel oriented, and therefore probably biased in this direction, the use of wind tunnel models, especially for dynamic testing is recommended without hesitation in the development of large-scale wind turbines.

#### SOME ITEMS FOR FUTURE RESEARCH

(a) Improved aerodynamic theory In comparison with HAWT's, the aerodynamic optimization of vertical axis turbines is still in its infancy. For example, simple symmetrical airfoils are generally used for blading. Unfortunately, further design refinements are hampered by inadequacies of available performance theories that are practical for curved-blade Darrieus type rotors. Momentum streamtube theory has not yet been developed to adequately predict the difference in induced velocities between the upwind and downwind forces of the rotor. Vortex theories have been developed which seem to be superior for the idealized two-dimensional (straight-bladed) rotor, but their adaptation to full three-dimensional flow is a formidable problem. In this state of affairs it is not even certain how to specify the most desirable airfoil characteristics. Low drag airfoil technology has not been much explored, but is potentially important because in constant-speed turbines parasite drag losses produce constant energy dissipation at all wind speeds.

(b) Torque and force filtering In two-bladed (or single-bladed) vertical axis turbines, there may be large-amplitude torque ripple, and also large-amplitude rotor drag and side force oscillations. Torque ripple can be reduced by designing the rotor and drive train so that the lowest natural torsional frequency is well below rotational frequency, but there are limits. Analysis of mechanical torque ripple filters has been carried out at NRC, and it is at least theoretically possible to reduce torque ripple to zero in constant-speed or nearly constant-speed systems. One method is to incorporate an elastically sprung flywheel in the drive train, with its natural frequency tuned to be equal to twice the rotor rotational frequency. The mass of the flywheel need only be about 2 percent of rotor mass. For turbines coupled to induction alternators, the operating speed varies by a few percent depending on power level, but there are simple bob-weight type "flywheels" that automatically retain their tuning over a range of shaft speed. These are old devices for torque smoothing in internal combustion piston engines. In the Maqdaen Islands turbine, no such devices are used, and in fact the magnitude of the torque ripple (roughly 20% of maximum mean torque) has posed

no apparent problems. Analysis also indicates that somewhat analogous bob-weight filters mounted near the top of the rotor, could be used to reduce or eliminate oscillatory drag and side force oscillations in the supporting guy cables.

(c) Soft mounts Two possible forms of soft mounts are shown in sketch form in Figure 3. Figure 3(a) shows a method of attachment of guy cables at the upper rotor bearing, which provides a low rotor whirl frequency while maintaining normal guy cable tension and natural frequencies. The sloping cables are attached to a floating ring and then continued vertically to outriggers on the bearing housing. The rotor natural frequency is a function of guy cable tension and the length of the vertical cable segments. The rotor will pass through its critical speed at a low rpm during starting or stopping but at normal operating speeds, the soft mount prevents significant oscillatory loads from being transmitted to the cables. This scheme has been investigated experimentally in wind tunnel tests of the 1/24 scale model of the Magdalen Islands turbine, with excellent results.

Figure 3(b) shows, in schematic form, a type of soft mount at the base of a VAWT. The design of fully cantilevered VAWT's normally requires a central column that has high bending stiffness, with consequent high costs. Depending on the relative mass distribution in the rotor and in the base, the addition of a soft base mount raises the bending natural frequency and therefore the effective stiffness of the rotor system. A low natural frequency of the entire assembly is also introduced but this can be damped relatively easily at the base (dampers not shown in the diagram). Further investigation of systems of this type may eventually make possible the cost effective development of large-scale VAWT's with no guy cable supports.

(d) Single blade rotors A 12ft (3.7m) diameter single-blade rotor was tested in the NRC V/STOL wind tunnel about 6 years ago, and was subsequently operated for some time outdoors. No serious problems were encountered, and performance was approximately equal to that of a multi-blade rotor of the same solidity. In late 1976, the 1/4 scale aeroelastic model of the Magdalen Islands turbine was modified by removing one of its blades and installing counter-weights. Wind tunnel tests showed no structural dynamic problems, and in particular no resonant condition corresponding to the two-bladed butterfly resonance. Cost estimates for large-scale turbines indicate that substantial cost reductions may be possible for single-blade rotors. Increase in blade solidity to the equivalent of a two-bladed rotor permits a lower blade material mass while retaining required blade stiffnesses. There is a possible drawback, however, and this may be true for single-blade horizontal axis turbines as well: they look strange,



particularly when rotating.

### CONCLUSION

The constant-speed curved-blade Darrieus type VAWT, of which the Magdalen Islands turbine is only one example, has been demonstrated at medium scale, and represents a potentially cost-effective WECS configuration because of its basic simplicity. It is still in its infancy, however, and can benefit from further research and development, provided always that cost reduction is kept as the main target.

### REFERENCES

1. R.S. Rangi, P. South and R.J. Templin, Wind Power and the Vertical-Axis Wind Turbine developed at the National Research Council - NRC DME/NAE Quarterly Bulletin 1974(2).
2. R.J. Templin and P. South. Canadian Wind Energy Program. Paper presented at the Vertical Axis Wind Turbine Technology Workshop. Held at Sandia Laboratories, Albuquerque, N.M., May 18 - 20, 1976.
3. P. South and A. Watts - Magdalen Islands VAWT Field Test. Paper presented at Workshop on Economical and Operational Requirements and Status of Large Scale Wind Systems, Monterey, Calif. 28 - 30 March 1979.
4. P. South, R. Rangi and R.J. Templin - Operating Experience with the Magdalen Islands Wind Turbine - Paper presented at the Second International Symposium on Wind Energy Systems, Oct. 3 - 6, 1978, Amsterdam.

## DISCUSSION

- Q. To what extent were your comments on the use of experimental models to determine dynamics effects affected by the fact that you have a relatively large wind tunnel to perform your experiments? On a commercial basis, analytical programs might be more cost effective, notwithstanding the fact that model demonstration really does provide the right answer.
- A. We were indeed lucky to have a large wind tunnel, and I also really didn't mean to imply that you should abandon analytical methods. There are advantages and disadvantages to each. The wind tunnel approach gives you an experimental answer for one design configuration, but it's much easier to change the variables in an analytical approach.
- Q. What have you experienced on the sensitivity of this type of rotor to gusts?
- A. If you refer to constant speed rotors so that the rotor inertia is not involved, they really are responsive to the gust scale that is of the same order of magnitude as the rotor dimensions. In that case, the power can fluctuate over a very wide range very rapidly.
- Q. Have you experienced extra stresses due to response of gusts in that situation?
- A. I don't really know whether I can answer that or not. We have taken blade stress data in the field and have put data through spectroanalysis. We find that the largest blade stresses are all at the precise multiples of speed where one finds them in non-turbulent flow. If there is a little spectral bump elsewhere that is not at a precise multiple of the speed, we think that is a signature of one of the rotor natural modes, such as the butterfly mode. It is probably occurring where it is because of random bumps due to, for example, turbulence or gusting. However, the loads are not high.
- Q. Can you define the geometry and the method of mounting the counterweight in a single-blade machine?
- A. In both of the models that I mentioned, the counterweights were just simple weights attached to the opposite end of the horizontal blade struts. They weren't mounted up at the ends.

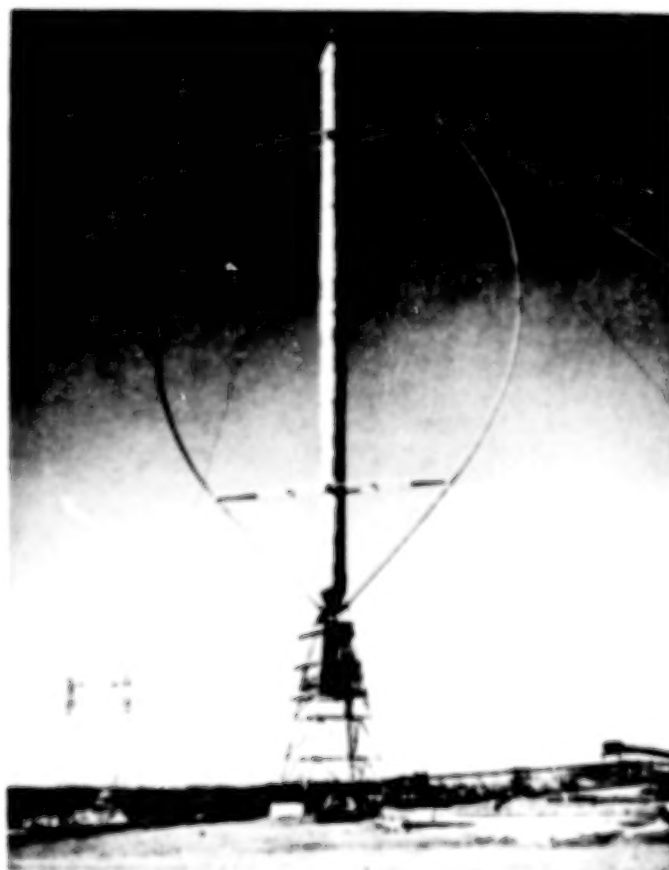


FIG. 1: 224 kW MAGDALEN ISLANDS VAWT.

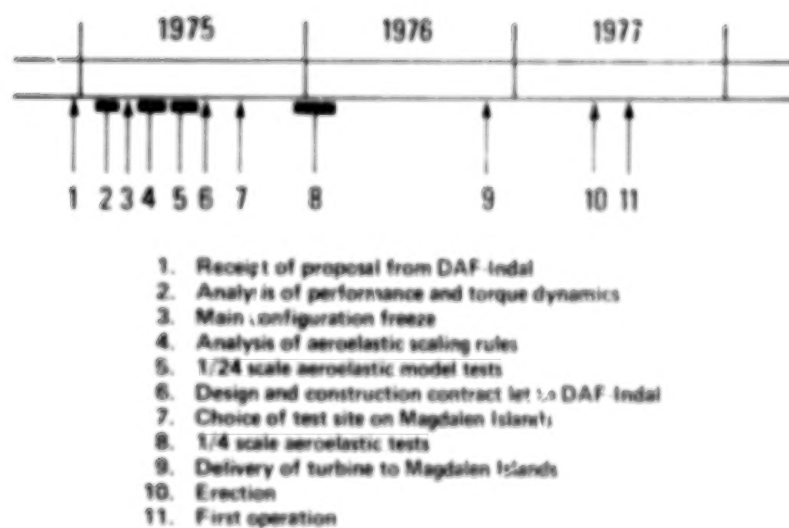
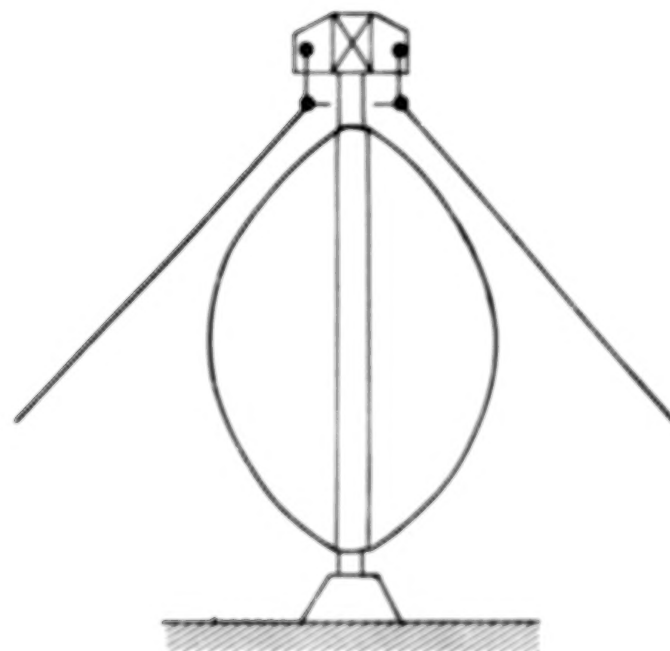
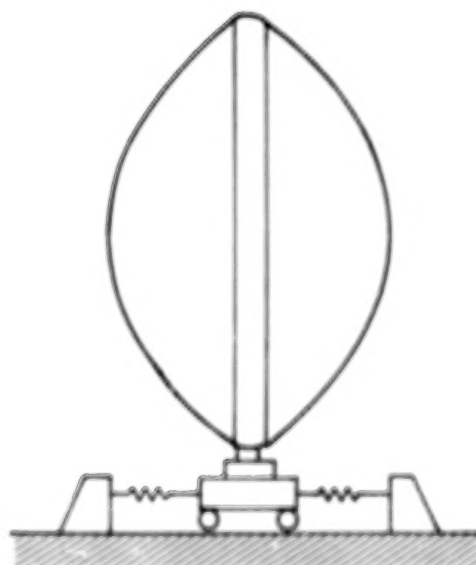


FIG. 2: DEVELOPMENT SCHEDULE - 224 kW MAGDALEN ISLANDS VAWT.



(a) Soft top mount



(b) Soft base mount

FIG. 3: SCHEMATIC SKETCHES OF TWO VAWT SOFT MOUNTS.

## ALCOA WIND TURBINES

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### OVERVIEW

The Alcoa wind energy program, initiated in 1975, began with the fabrication of turbine blades, and two years later broadened its scope to the design and fabrication of complete systems of Darrieus-type vertical-axis wind turbine.

Alcoa's interest in blade fabrication is natural, as the extrusion process is uniquely suited to the production of all-aluminum blades with constant chord. Since A. G. Craig, Jr. is going to present "Fabrication of Extruded Vertical Axis Turbine Blades" in the same Workshop, the discussion on blades here will be brief.

1. Alcoa served as a subcontractor to Kaman Aerospace in the fabrication of the original 21-inch chord blades for Sandia's 17-m research Darrieus turbine.<sup>1</sup> The blade is of a helicopter blade design and each blade has five sections with two or them forming V-shaped struts as supports.
2. Alcoa fabricated the 6-inch chord blades for Sandia's 5-m research turbine<sup>2</sup>, and the same blade has been used for many other machines including the Dynergy 5-m and Clarkson College's silo-mounted turbines.
3. Alcoa extruded the new 24-inch chord blades and formed them at the site for the Sandia 17-m. The blades are continuous 80-foot long pieces without joints or struts.<sup>3</sup> The same blade will be used for the Low Cost 17-m machine funded by the U.S. Department of Energy.
4. Alcoa has extruded a 14-inch chord blade for a 60 kW turbine and a 29-inch chord blade for a turbine with a rated power in excess of one quarter MW.
5. Blades with chord up to 58 inches, large enough for MW size machines, are under study.

6. In addition to blades for Darrieus turbines, Alcoa has also supplied a 24-inch chord blade with the cross section of an unsymmetrical airfoil to Grumman Aerospace for the Wind Stream 25, a horizontal axis wind turbine with a rated power of 15 kW.

In the area of complete systems, a family of five Darrieus turbines named ALVAWT (Alcoa Vertical-Axis Wind Turbine) are in demonstration runs within a year. All systems are designed to operate as constant rpm devices to be interfaced with utility grids to produce AC 3-phase at 60 Hz.

In figure 1, the schematic of the basic ALVAWT models are shown. The number system adopted designates the rotor size, with the first two digits\* indicating the rotor height in feet, the second two digits the rotor diameter in feet, and the last two digits the chord of the blade in inches. All models show the 2-bladed rotor with single torque tube supported on top with three tiedown cables.

The development of the second and the fourth turbines, 453011 and 835524, is funded by the U.S. Department of Energy. Number 453011 is also known as the 8 kW VAWT in the Rockwell International's Rocky Flats program, and 835524 is also known as the Low Cost 17-m, which has received technical input from Sandia. The Low Cost 17-m represents the central effort at Alcoa to date, and the other models in the ALVAWT are simply either scale-down or scale-up versions of this machine. A detailed description of the Low Cost 17-m is given in the following section.

A variation of the 271806 may also be of interest as it materialized as the result of a joint venture initiated by Clarkson College of Potsdam, New York.<sup>4</sup> This turbine, installed at Clarkson College, has many unique features; it is mounted high on the side of a silo with no cable support; it has a continuously varying ratio speed increaser; and it is equipped with the flexibility of either a 15-foot height/diameter ratio of 1.0 or an 18-foot height/diameter ratio of 1.5 rotor, with each rotor having either two or three 6-inch blades. The system is a research vehicle designed for rural application where silos are readily available. Its free-standing design also suggests the possibility that it could be installed on top of tall buildings.

The 634214 is a machine in an overlapping size which can be considered as a small system for rural or residential use,

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\*The exception is the fifth or the largest model which needs three digits.

but in the meantime possessing all the characteristics of a utility machine. It is still small enough to be shipped by conventional trucks and installed easily. Several units have been planned for delivery, with the first one scheduled to be installed in western Pennsylvania in the fall of 1979.

The largest model in the ALVAWT family is the 1238229 which is an enlarged version of the Low Cost 17-m using the 29-inch chord blades. If installed today, it would be the largest VAWT in the world. This machine is intended for utility application at high wind sites. It could be rated at 1/2 MW for a site with an annual mean wind speed of 8.05 m/s (18 mph). A prototype is scheduled to be installed for testing at the Alcoa Technical Center in Alcoa Center, Pa., in early 1980.

The performance specifications, prepared with the aid of Sandia's mathematical model PAREP,<sup>5</sup> are presented in Table 1.

Table 1  
ALVAWT PERFORMANCE SPECIFICATIONS  
(Annual Mean Wind Speed 15 mph)

Model	Rated Power kW	Rated Wind Speed m/s (mph)	Turbine Speed rpm	Generator hp	Annual Energy Output MW-Hr
271806	8	15.2 (34)	164	10	14
453011	26	14.8 (33)	96	30	52
634214	55	14.8 (33)	72	75	109
835524	114	13.9 (31)	52	150	246
1238229	280	13.0 (29)	34	400	625

In addition to the five basic models, Alcoa Laboratories is also studying a MW unit. It has been recognized that large Darrieus VAWT systems can be installed economically.<sup>6,7</sup> However, a fair amount of research and development is believed to be needed before MW units can be readily fabricated.

#### THE LOW COST 17-m

As part of the Federal Wind Energy Program to accelerate the development, commercialization and utilization of reliable and economically viable wind energy conversion systems, the program "Design and Fabrication of a Low-Cost Darrieus Vertical-Axis Wind Turbine System" was initiated in 1978. The resultant turbine is 17 meters in rotor diameter, producing 100 kW AC electrical power and compatible with a utility grid. The Sandia 17-m was used as the background machine from which design information was drawn, and the U.S. Department of Energy provided technical support to the program through Sandia Laboratories. The new machine is referred to as the "Low Cost 17-m", or 835524 if the Alcoa number system is used.



A major objective of this program is to obtain realistic fabrication cost data, based on current technology, with the goal of minimizing costs of energy generated. Another objective of this program is to provide a low-cost system design suitable for continued production and/or to serve as a base line for further cost reduction efforts.

The program has two distinct phases: Phase I was a seven-month design phase. Its objective was to produce detailed design layouts and drawings for fabrication and is now complete. Phase II is a fabrication and installation phase to utilize the design completed in Phase I. Four units have been authorized for demonstration by the U.S. Department of Energy, with the first unit to be located at the small Wind System Test Center at Rocky Flats, CO. Work on Phase II is expected to begin shortly.

#### System Definition

The Low Cost 17-m is similar to the Sandia 17-m research turbine with certain configurational changes aimed at cost reduction. By concentrating the modifications on an existing design, emphasis was focused on component cost reduction rather than selection of optimal configuration or operating modes.

The Low Cost 17-m has two blades. Each blade has a uniform cross-section of a 610 mm (24 in.) chord NACA 0015 airfoil (Fig. 2). For practical fabrication reasons the nominal blade shape is the straight-circular straight approximation to the troposkien as in the case of the Sandia 17-m. However, a height/diameter ratio of approximately 1.5 has been selected to give the new machine 50% more swept area. The blade for the new machine is the same as the current Sandia 17-m and no strut is required.

The torque tube or tower is a single, spiral-welded steel tube of 762 mm (30 in.) diameter supported at the top bearing by three tiedown cables and at the bottom bearing by a simple steel frame resting on a concrete foundation. The blades are attached to the torque tube by means of truss-like connectors with the lower ends of the blades located 3.048m (10 ft) above ground. The two blades and the torque tube form the rotor which is the rotating part of the system.

For comparison, the geometrical configuration parameters of the Low Cost and the Sandia 17-m are presented in Table 2.



Table 2 Configuration Parameters

	<u>Low Cost 17-m</u>	<u>Sandia 17-m</u>
Rotor Diameter	17 m (55 ft)	17 m (55 ft)
Rotor Height	25.15 m (82.5 ft)	17 m (55 ft)
Ground Clearance	3.048 m (10 ft)	4.27 m (14 ft)
No. of Blades	Two	Two or Three
Blade Cross Section	NACA 0015	NACA 0012 (O),* NACA 0015 (N) **
Blade Chord	610 mm (24 in.)	533 mm (21 in.,O),* 610 mm (24 in.,N) **
Blade Length	30.48 m (100 ft)	24 m (79 ft)
Nominal Blade Shape	Straight-Circular-Straight	Straight-Circular-Straight
No. of Struts	None	Two per Blade (V-Shaped,O),* None (N)**
Rotor Type	Single Tube	Single Tube
Tube Size	762 mm (30 in.) O.D. 5.6 mm (0.219 in.) wall	0.5 m (20 in.) O.D. 25.4 mm (1 in.) wall
Rotor Support	Guyed on Top	Guyed on Top
No. of Cables	Three	Four
Cable Size	22.2 mm (0.875 in.)	25.4 mm (1 in.)
Cable Angle	35°	35°
Cable Length	50 m (164 ft)	39.3 m (129 ft)
Overall Height	28.9 m (94.9 ft)	28.65 m (94 ft)

\* O = Original blade (Kaman)

\*\*N = New blade (Alcoa)

The total weight of the system, excluding the concrete foundation for the main structure and footings for the tie-down cables, is approximately under 12,272 kg (27,000 lb). It is believed that further reduction in weight is possible when fabrication techniques are refined in Phase II of the project. As a comparison, the Sandia 17-m has a total weight of 16,800 kg (37,000 lb).

The system has a simple drive train which is defined as the series of components to transmit and to convert the aerodynamic torque from the rotor to electrical power delivered to the utility grid. It begins with the rotor which is connected by a low-speed shaft to a speed increaser, and the speed increaser is connected by a high-speed shaft to a relatively high-speed induction motor. The induction motor is directly connected to a utility line and maintains the drive train at a nearly constant speed, corresponding to its synchronous speed. The induction motor acts as a generator when wind-produced torques from the rotor minus all losses in the drive train are positive. The same unit acts as a motor to start the turbine from rest. The drive train

is provided with a braking system mounted on the high-speed shaft to stop the turbine when the induction motor/generator is decoupled from the utility line. The schematic of the drive train is shown in Fig. 3.

The nominal rotational speed of the turbine is selected as 51.5 rpm corresponding to the selection of a rated electric power of 100 kW for the system. The rated power would be realized when the wind speed reaches 13.86 m/s (31 mph) measured 9.1 m (30 ft) from the ground. The induction motor chosen for the system is a 150 hp, 1800 rpm unit with a generator slip of 1.1%. The nominal rotational speed of the turbine and the synchronous speed of the induction motor determine the fixed ratio (35.068) of the speed increaser. In the starting mode, the induction motor is capable of providing sufficient torque to overcome losses in the system in order to accelerate the rotor to 51.5 rpm in less than 15 seconds.

#### System Operations

Two kinds of operation of the system have been identified. The first kind is the normal operation which is defined as the normal power generation of the turbine between a cut-in wind speed of 5.81 m/s (13 mph) and a shut-down wind speed of 26.8 m/s (60 mph) at a turbine speed of 51.5 rpm. Since wind is fluctuating by nature, the wind speeds selected for cut-in and shut-down are averages over pre-set time intervals.

The second kind is the emergency shut-down operation triggered by certain anomalies which might lead to the runaway condition. The turbine structure is designed with adequate strength against an arbitrarily-determined runaway condition of a combined situation of a turbine speed of 75 rpm and a wind speed of 33.5 m/s (80 mph). For the turbine to accelerate from its nominal speed of 51.5 rpm to 75 rpm, a five-second interval exists allowing the system to activate the emergency brake.

A controller is designed to provide both fail-safe automatic operation or manual operation of the system. Either mode is selectable by means of panel-mounted selector switches.

In automatic mode the controller, sensing wind speed via an anemometer, will automatically start the turbine when the cut-in wind speed is passed. Conversely, when wind speed falls below the cut-in speed, the controller will automatically shut down the turbine. The controller will also shut down the turbine when the shut-down speed is reached.

In manual mode the turbine may be started independently of the anemometer by sequential operation of the brake release and turbine start push buttons. The anemometer will, however, provide for shut-down wind speed protection as in automatic mode.

In order to avoid possible damage to the system, many safety features have been incorporated via the controller. The system will be shut down if any one of the following abnormal conditions is encountered: excessive vibration, overspeed, loss of line, ground fault, and computer program failure.

The most cost effective way of implementing this controller, providing for the protection and proper sequencing of the brakes, starter and other subsystems, has been determined to be a micro-computer. A schematic of the control diagram is shown in Fig. 4.

To summarize, parameters associated with the operations are presented in Table 3 for the Low Cost 17-m and the Sandia 17-m.

Table 3. Operation Parameters

	<u>Low Cost 17-m</u>	<u>Sandia 17-m (Original)</u>
Rated Electric Power	100 kW	60 kW (3-Blade) 50 kW (2-Blade)
Generator Capacity	112 kW (150 hp)	60 kW
Rated Wind Speed	13.86 m/s (31 mph)	14.3 m/s (32 mph)
Cut-In Wind Speed	5.8 m/s (13 mph)	--
Shut-Down Wind Speed	26.8 m/s (60 mph)	26.8 m/s (60 mph)
Nominal Turbine rpm	51.5	52.0
Designed Max. Allowable Turbine rpm	75.0	75.0
Nominal Turbine Torque	21 130 N·m (15,585 ft-lb)	13 558 N·m (10,000 ft-lb)
Nominal Brake Torque	59 656 N·m (44,000 ft-lb)	67 790 N·m (50,000 ft-lb)
Emergency Brake Torque	59 656 N·m (44,000 ft-lb)	67 790 N·m (50,000 ft-lb)

All wind speeds in Table 3 are measured 9.1 m (30 ft) from the ground.

#### General Design Criteria

The major turbine and drive train components are designed for a productive life of approximately 30 years. All portions of the system can withstand weather extremes typical of the U.S. and protection is provided from the weather for all delicate components. The main structure is designed to survive a maximum wind speed of 67.1 m/s measured at 9.1 m above ground (150 mph, 30 ft). The system is also more than adequate to survive earthquake loads for seismic zone 3.

## Performance Analysis

The sole function of a wind turbine is to convert kinetic energy in the ambient air stream into usable energy. As such, the performance analysis of the wind turbine system is, in essence, the determination of the power coefficient,  $C_p$ , under a wide range of wind conditions, and the determination of the total electric energy output per year depending on site characteristics. These calculations were carried out with the aid of computer programs developed by Sandia Laboratories.

For the calculation of  $C_p$ , the program "PAREP" was again supplied. For the Low Cost 17-m operating at its normal turbine rotational speed of 51.5 rpm, the  $C_p$  plotted against  $\lambda$ , the rotor tip speed/ambient wind speed, is presented in Fig. 5.

The electric power output versus ambient wind speed measured at the center of the rotor is shown in Fig. 6. The peak of the curve represents the rated power of the turbine system. The wind speed at the center of the rotor is in general higher than that measured 9.1 m (30 ft) from the ground due to the added height.

The total energy output of a turbine system is a strong function of its site. In order to assess the total energy output in a year, the wind distribution data at the site must be known. The Sandia program, VERS 16, was used for the total energy output calculations using typical annual wind duration curves as shown in Fig. 7.

The total energy production per year of the Low Cost 17-m for an annual mean wind speed of 6.71 m/s (15 mph) was computed as 235,000 kW·h/yr. As a comparison, the 3-bladed Sandia 17-m at the same site would produce a total energy output of 160,000 kW·h.

## Cost Estimate

As mentioned earlier, the major objective of this program is to obtain realistic fabrication cost data based on current technology. However, costs are known to be sensitive to volume of production. One can readily see that it is highly unlikely that the same manufacturing process would be applied to a production of a few demonstration units as to a production of, say, one hundred units. To go one step further, the 100th unit cost of a production of 100 units within a certain period of time on a continuous base may again differ considerably from the 100th unit cost in a single order of 100 units. It is apparent that by constructing different scenarios, one may come up with just as many different cost estimates.

For the case of the Low Cost 17-m, the assumption made is that the unit cost is defined as the average cost per unit based on a production of 100 units per year on a continuous schedule of approximately two units per week. The facility is a new "Greenfield" plant optimized for the production for intermediate sized vertical-axis wind turbines typified by the Low Cost 17-m based on a manufacturing process recommended by Alcoa's Allied Products Department after reviewing the design completed in Phase I. The numbers generated in this scenario would be considered highly probable rather than optimal. It should be emphasized that all figures are based on 1978 October dollars and must be adjusted for inflation to whatever time period that production will be required or costs are to be compared to other alternatives.

The "Greenfield" plant is sized at approximately 25,000 sq. ft. and staffed with six people in administration, engineering and sales, and twelve management, clerical, and support people as part of the overhead.

For direct cost estimates, it has been assumed that adequate labor will be available at an average cost of \$5.00 per work hour and that a 30% payroll benefits package will apply. Those figures are approximately median in the sixteen subsidiaries that compose the Alcoa Allied Products Department. Furthermore, a single-shift, five day schedule in a fifty week work year has been assumed.

Preliminary production plans show a need for 615 person hours for fabrication and assembly, which suggest a need for 32 direct production workers and, with allowances for vacations and holidays, an annual payroll of \$433,000 for 61,500 hours of productive work. Projections of annual material and component purchases total \$3,842,000. The component cost and direct labor per unit are shown in Table 4.

Table 4. Direct Material and Labor Cost

	<u>Person Hours</u>	<u>Material Cost</u>
Rotor Components	345	\$ 12,800
Base Assembly/Mechanical/ Electrical Subsystems	262	\$ 24,495
Miscellaneous Components	8	\$ 1,125
	<u>615</u>	<u>\$ 38,420</u>
Direct Labor Cost		<u>\$ 4,330</u>
Direct Material/Labor Cost		\$ 42,750



Adding corporate and production overhead, the total production cost of 100 units per year is given in Table 5.

Table 5. Total Production Cost - 100 Units/Year

Direct Materials Purchased	\$ 3,842,000
Direct Labor and Benefits	433,000
Production Overhead	536,000
Corporate Overhead	<u>472,000</u>
Total	\$ 5,283,000

Assuming a gross margin goal that will result in a 25% return on capital in use, and capital requirements of \$1,180,000, the annual profit goal is \$295,000. Therefore, total revenue is projected at \$5,578,000 and an average selling price of approximately \$56,000 per unit is dictated.

Site preparation, including three tiedown footings and a base foundation, is estimated to cost approximately \$5,000 if ground conditions are "average." Delivery costs, with average shipping distances of approximately 250 miles, would add another \$500 to the cost; and unloading, assembly, erection, hookup and checkout of the system should cost an additional \$15,000-\$19,000.

The total installed cost would, therefore, be \$76,500-\$80,500 depending on the site related cost. With experience, design refinement, increased confidence levels and backward integration into more production operations (as opposed to purchased components), an additional cost reduction of approximately \$11,000 is believed possible in an established business enterprise.

Therefore, the total installed cost of the Low Cost 17-m, ready for delivery of three-phase 60 Hz AC electricity at 460 volts, should be in the \$65,500-\$80,500 range in 100 unit annual production quantities.<sup>8</sup>

The cost of energy to the owner is calculated based on the following formula:

$$COE = \frac{\left( \frac{\text{Total installed cost}}{\text{cost}} \right) \times \left( \frac{\text{Annual charge}}{\text{rate}} \right) + \left( \frac{O\&M}{\text{yr}} \right) \times \left( \frac{\text{Levelization}}{\text{factor}} \right)}{\text{Total electric energy output/yr} \times \text{Availability factor}}$$

In the calculations, three annual charge rates, 12, 15 and 18%, have been assumed to cover the spread to different owners because of interest rate difference, tax structure, or other reasons. The O & M per year is chosen as \$773.00 which is the number used by Sandia for the Sandia 17-m. The levelization factor is taken as 2.0. In the denominator, the figures 235,000 kW·h and 0.9 are used as the total electric energy output per year and the availability factor, respectively. The results, therefore, represent the cost of energy at a site with a 6.71 m/s (15 mph) annual mean wind speed.

Table 6. Cost of Energy

<u>Total Installed Cost</u>	<u>Annual Charge Rate</u>	<u>\$/kW·h</u>
\$ 65,500	12%	0.044
80,500		0.053
65,500	15%	0.054
80,500		0.064
65,500	18%	0.063
80,500		0.076

Numbers in Table 6 suggest that the competitiveness of wind energy becomes highly dependent on the annual charge rate and the availability and cost of alternative forms of energy. This indicates that for the same site one type of owner may favor wind energy while another type may not.

#### REFERENCES

1. Barzda, J. J., "Blade Design and Fabrication of 17-m VAWT," Vertical-Axis Wind Turbine Technology Workshop, pp. II 205-221, May 17-20, 1976.
2. Ai, D. K., "A Low Cost Blade Design for a Darrieus Type Vertical-Axis Wind Turbine," Wind Technology Journal, Vol. 2, Nos. 1 & 2/Spring and Summer, pp. 18-23, 1978.
3. Craig, A. G., Jr., "Fabrication of Extruded Vertical Axis Turbine Blades," Proceedings, Workshop on Large Wind Turbine Design Characteristics, NASA-LeRC, Cleveland, Ohio, April 24-26, 1979.

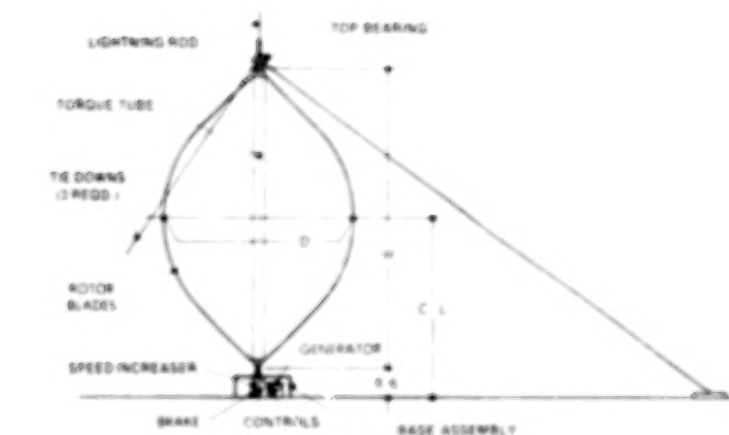
4. Rollins, J. P., and Kear, E. B., Jr., "Progress Report on Silo-Supported Darrieus Wind Turbine," presented at the Fall Meeting of American Wind Energy Association, Hyannis, Mass., September 1978.
5. Klimas, P. C., and Sheldahl, R. E., "Four Aerodynamic Prediction Schemes for Vertical-Axis Wind Turbines," A Compendium, Sandia Laboratories, SAND 78-0014, June 1978.
6. "Parametric Optimization Study - Darrieus Type Vertical Axis Wind Turbines," Phase II, Final Report, Alcoa Laboratories. Prepared for Sandia Laboratories, August 31, 1978.
7. Sullivan, W. N., "Economic Overview of Vertical Axis Wind Turbines," EPRI/DOE Workshop on Economic and Operational Requirements and Status of Large Scale Wind Systems, Monterey, California, March 28-30, 1979.
8. Ai, D. K., et al., "Design and Fabrication of a Low Cost Darrieus Vertical-Axis Wind Turbine System, Phase One Technical Report," U.S. Department of Energy, Contract No. EM-78-C-04-4272, 1979, to be released.

#### DISCUSSION

- Q. What aluminum material was used in the extrusion, and also what protection system, if any, was used?
- A. The extrusion material can either be 6063 or 6061. We have done both. As far as protection is concerned, I believe the Sandia blades are painted. We haven't done anything in this regard, and we haven't really addressed the problem. If it's necessary, say if the blade is to be used in a corrosion environment, then we will provide the proper finish. The blades may even be Teflon coated, if there is concern about ice. We have a lot of capabilities in the area of surface finishing.
- Q. Are you prepared to supply one of these units at the price of \$100,000?
- A. Yes, on this particular design we will deliver the first four units to DOE. The next one will then be available. In fact, our subsidiary is very anxious to start. They really believe it is a viable commercial item at that price.
- Q. Have you included land requirements and loan costs?
- A. Loan costs, yes, but not land requirements.



- Q. Can you discuss the flexibility that is incorporated into the drivetrain?
- A. The main point is to have the torque ripple stay within a certain limit. Ours is less than 20 percent. The result is based on a theoretical model developed by Sandia. The drivetrain is softened up in order to do that by having an adjustable low speed shaft, etc.
- Q. What is the slip of your induction generators?
- A. The slip was 1.1 percent, which means the unit is operating between 1,780 and 1,820 rpm.
- Q. One of the things discussed was thrust bearings. A thrust bearing is carrying the total weight of the turbine in your installation. The thrust bearing can be a difficult problem since it carries the total weight and is involved in starting, stopping and lubrication. What do you do?
- A. That was a problem we have given a lot of thought to. We have a lower bearing supported by a steel frame, and the load is transmitted to the frame. With this type of design with guy wires, the lower thrust bearing carries the weight of the rotor plus the vertical component of the tension from the guy wires. The bearings will be running in oil. We have done our calculations and selected the bearing to meet the requirements. The bearing should be adequate according to its specification.
- Q. What is the Alcoa business purpose? Are you to sell vertical axis machines and extruded blades or just what?
- A. Our business purpose is two-fold. We are interested in selling blades because that is a mill product. We are in that business. We are also interested in energy conservation and production, and more and more it looks like this is a diversified business.
- Q. For the 24-inch extrusion, where is the Alcoa press, and what does a recurring extrusion die cost if a design is different than what you have?
- A. There are four 14,000 ton presses in the western world. Alcoa owns two of them. Both are located at Lafayette, Indiana. The die cost for the 24-inch or 29-inch blade is approximately \$20,000. The reason that the prices are similar is because they fit into the same sized cylinder. It should be noted that tooling cost relates to the chord of the blade and not to its length. That is an important consideration for making the blade inexpensively.



**TYPICAL ALVAWT  
ALCOA VERTICAL AXIS WIND TURBINE**

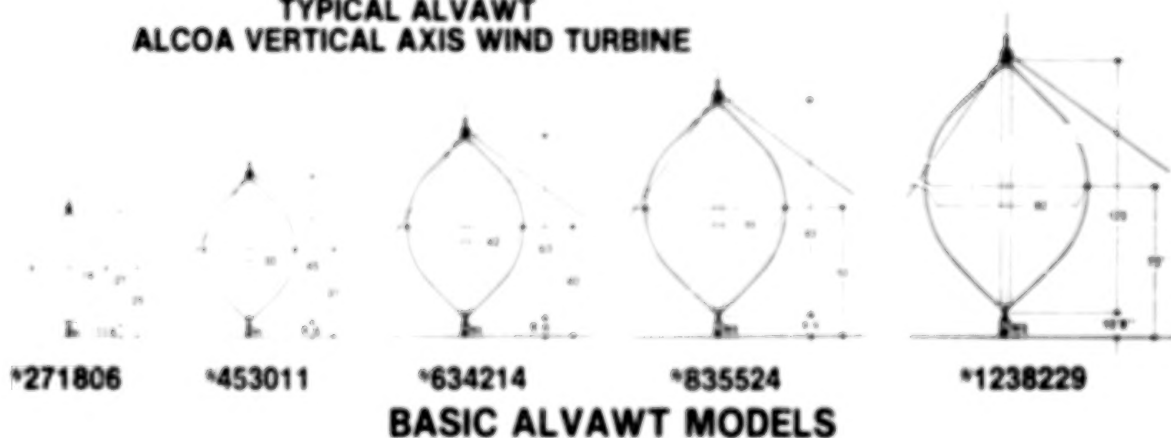
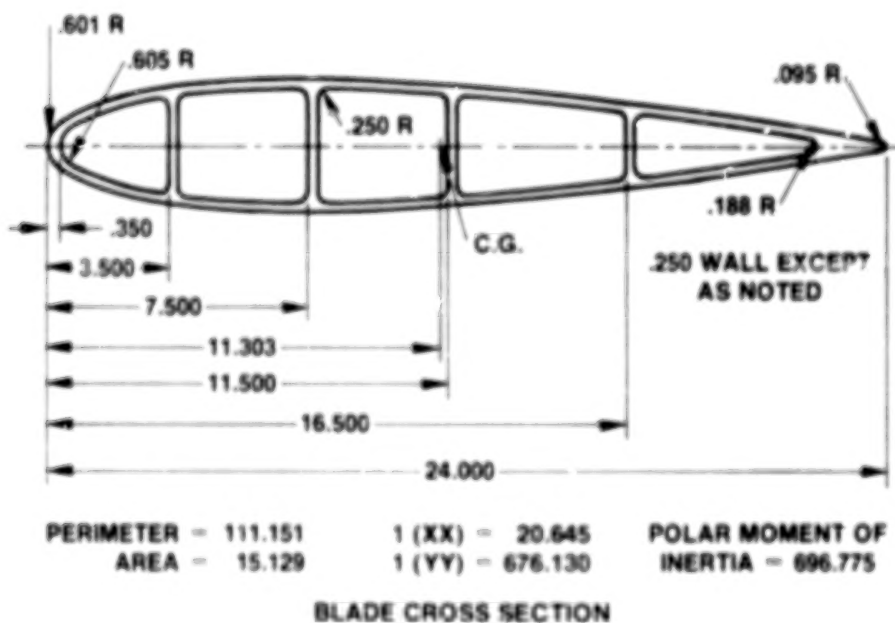


Figure 1



**THE EXTRUDED 24-INCH CHORD ALUMINUM BLADE**

Figure 2

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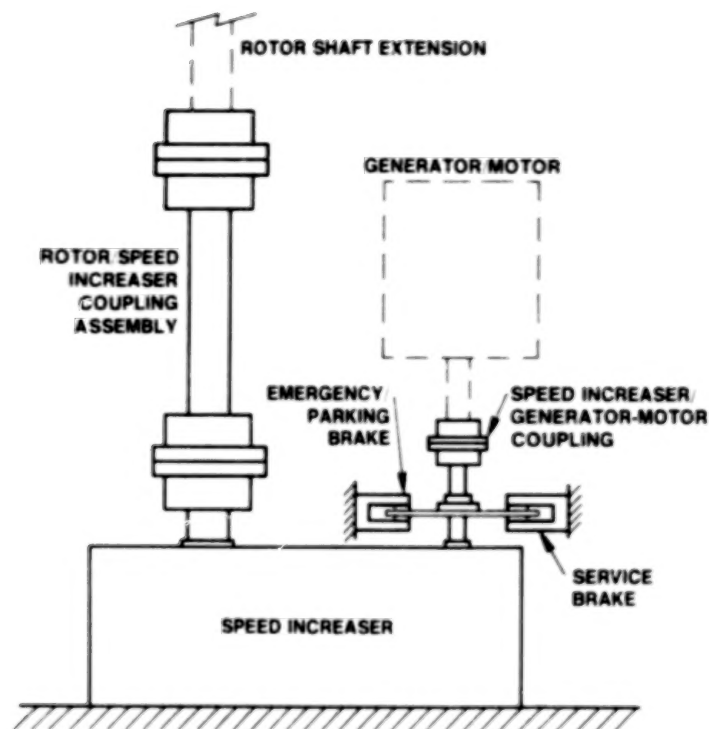
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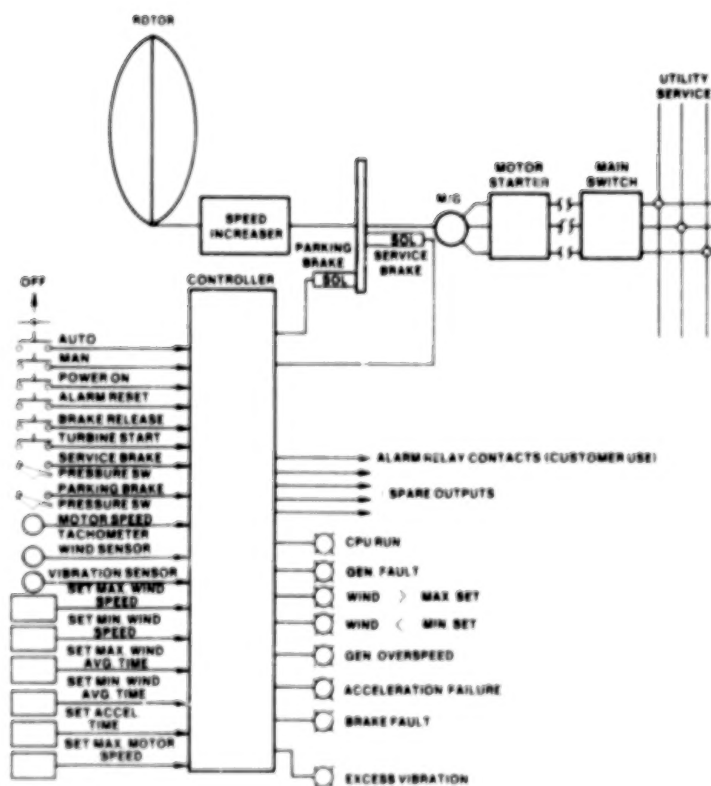
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DRIVE TRAIN SCHEMATIC

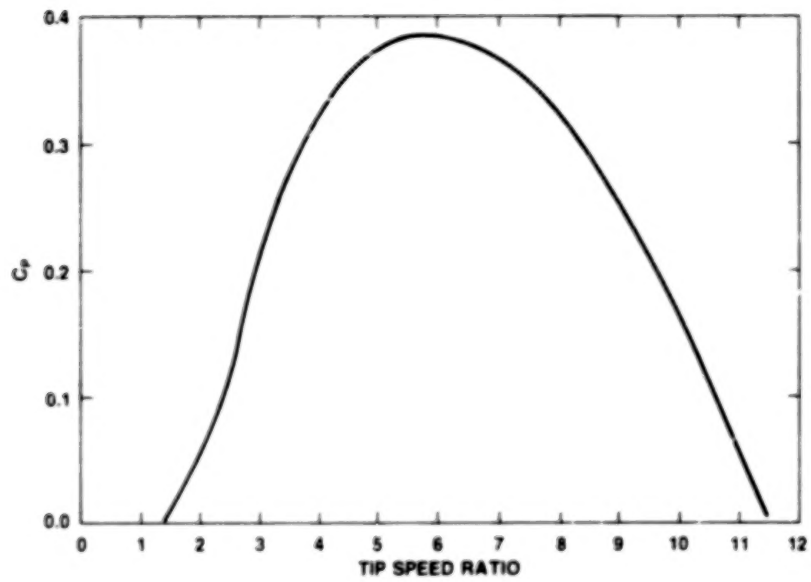
Figure 3



SCHEMATIC OF CONTROL

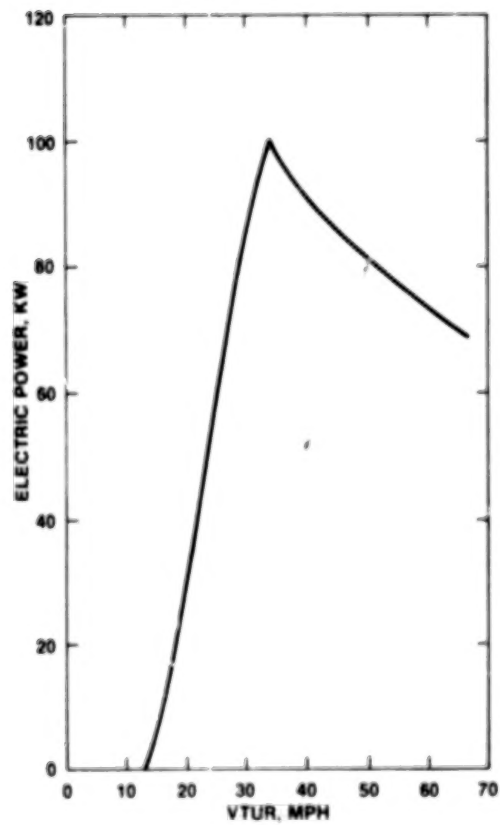
Figure 4

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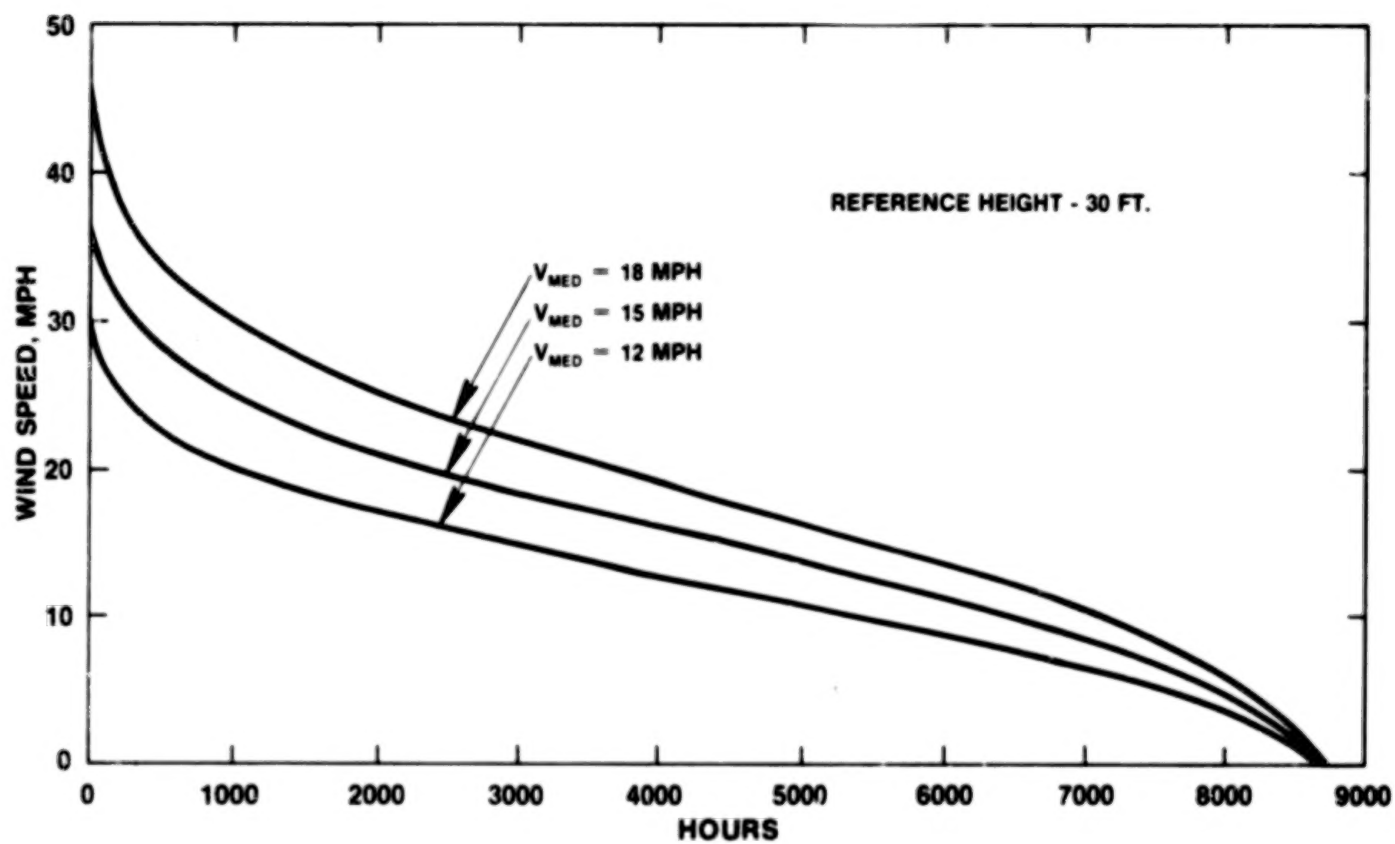
**TURBINE RPM = 51.5**

Figure 5



**TURBINE RPM = 51.5**

Figure 6



### ANNUAL WIND SPEED DURATION CURVES

Figure 7



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## TEST RESULTS OF THE DOE/SANDIA 17 METER VAWT\*

Robert O. Nellums and M. H. Worstell  
Sandia Laboratories  
Albuquerque, New Mexico 87185

The DOE/Sandia 17 meter wind turbine began collecting operational data on March 29, 1977. Since then, operation has been conducted on a variety of rotor and drive train configurations. The present log of over 530 hours is an important means of validating theoretical calculations and providing operating experience. This report will contain a brief review of the test program followed by a presentation of the performance results and their significance. Then, in order to provide the reader with an appreciation of the 17 meter operating experience, this report will close with a discussion of the operational difficulties occurring since the first turn 2 years ago.

### The 17 Meter Test Program

The characteristics of the 17 meter turbine have been summarized in table I, and the present turbine configuration is illustrated in Fig. 1. Of particular significance is the operation of the turbine in a synchronous mode with the power grid. Control of the turbine is accomplished manually requiring the presence of an operator.

The turbine has been heavily instrumented for data collection. Windspeed is measured by two anemometers situated on a tower 22 feet above the rotor. This measurement may be correlated by recordings at four heights on a nearby tower. The anemometers used are Teledyne Geotech Model 1564B with specified accuracy of  $\pm 1\%$ . Windspeed is corrected to centerline and 30 foot height according to a 0.1 shear factor which has been determined experimentally for the site<sup>1</sup>. The measurement of windspeed is a critical function which has received great attention.

The power train is instrumented at several locations. Rotor aerodynamic power is measured via a precision torque sensor on the low speed shaft. The measurement of this torque sensor is corrected for bearing loss which has been experimentally determined to be 287 ft-lb at standard test conditions. A second torque sensor is mounted on the high speed shaft, permitting transmission loss to be determined. RPM, electrical output voltage, current, and power are measured to complete the power train measurements.

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\*This work prepared for the U.S. Department of Energy, DOE, under contract DE-AC04-76DP00789.

Stress levels in the system are measured at several points. The brake and transmission temperature are measured. Multiple strain gages generate tower and blade stress information which is pulse code modulated and transmitted to recording instrumentation through a slip ring.

The data collected by these instruments are processed according to the "method of bins"<sup>2</sup>. By this means, average values of power, torque, etc. are calculated as a function of windspeed. This method has been found to yield highly repeatable results for the 17 meter test system; the data presented in this report are based on summing all available data into the calculation.

The various blade and power train configurations which have been tested are shown in table II. The test program has encompassed variations in blade number, blade shape, transmission type, and induction motor size.

### Performance Results

Prior to February 1979, testing of the 17 meter turbine was conducted using 21 inch, NACA 0012 blades combined with support struts. The performance testing of these blades has been documented<sup>3</sup>. In March of 1979, testing commenced using 24 inch, NACA 0015 blades without the use of struts. Several preliminary results for the new blades will be presented and compared to the old blades. The results reflect the air density in Albuquerque of .0625 lbm/ft<sup>3</sup> and have not been corrected for sea level.

Selected test results are shown in Figs. 3 to 8. Special attention should be directed to Fig. 8. These preliminary results for the new blade indicate an improvement in performance beyond expectation. Whereas the former strutted configuration performed below analytic prediction, the present unstrutted configuration is exceeding forecast efficiencies over portions of the windspeed range. The observed efficiencies are very favorable.

Analytic calculations in the past have indicated that Darrieus turbines such as the 17 meter test turbine are inherently less efficient than horizontal wind turbines. However, a peak efficiency of 40.9% has been measured which is believed to be comparable to any horizontal axis experience to date. It is hoped that future experimental data will help to clarify the relative efficiency of the Darrieus concept.

### 17 Meter Test Turbine Encounters with Problems

The inclusion of operating difficulties in this report has not been motivated by the existence of large problems. On the contrary, it is hoped that inclusion will serve to highlight an unusually favorable record for a new concept prototype. The test program to date has not uncovered a single problem likely to affect the economic viability of the Darrieus turbine. Most of the problems involve test instrumentation

not pertinent to normal machine operation. Of the remainder, it is believed that by identifying potential pitfalls here, future designers may be able to avoid them. The difficulties which have been experienced are shown in Fig. 9.

The first problem discovered during testing was an inherent high power loss in the transmission reaching 30 kW. This condition also aggravated a tendency of the 50 hp induction motor to overload during start up. These problems were addressed by first disassembling and inspecting the transmission. After finding no faults, the wet sump lubrication was replaced with a dry sump system. Following this modification, the transmission efficiency improved dramatically as is shown in Fig. 3. No additional start up problems have arisen. A final modification at this time was to increase the induction motor to 75 hp in order to expand the system generating capability using an induction generator. The rating increase was also expected to complement the start up capability.

Occurring next in chronological order were several faults in the data collection instrumentation which had no bearing on normal machine operation. The pressure transducer used to indicate brake pressure failed and was replaced with no problems since. The Pulse Code Modulator used to transmit multiple strain gage measurements developed a faulty power supply which was replaced and no problems have resulted since. The torque sensor used to measure rotor output suffered water damage and had to be rebuilt. Last, the LED readout of RPM became erratic and was replaced.

The lightning protection system for the 17 meter test system consists of a top mounted mast connected to ground through the four guy cables and through the tower via dedicated slip rings<sup>4</sup>. Only the mechanical system is protected. The turbine has undergone five lightning strikes; only the last strike caused discernable damage. High voltage passed through the anemometer output wires into the computer interface, damaging several circuit boards. Future effort will be aimed at protecting electronic components.

The most recent difficulty to arise has been the loosening of several bolts and drive shaft keys. As a result, regular inspection of bolts and keys is now performed. The keys were only found to be loosening on shafts where the key was the only locking mechanism transmitting torque. These keys are being replaced by clamp-type arrangements. Both of these problems are to some degree associated with the oscillatory output of the turbine which should be reduced on future designs as a result of torque ripple reduction studies<sup>5</sup>.

Two final problems are minor but have been consistently bothersome. The blade hinge pins tend to seize to their bushings and complicate the changing of blades. Secondly, the anemometers are extremely subject to damage from hailstorms, ice, and high wind and have required frequent

repair. Several more robust units are being investigated for use in measuring cut-in and cut-out windspeed such as might be done on a commercial design. However, the test program requires the finest and most precise windspeed measurement available and it appears that the current rate of repair may be unavoidable on the 17 meter testing.

To summarize the conclusions of this report, the 17 meter diameter test bed has thus far produced power efficiency and reliability experience favorable to the Darrieus turbine concept. Continued experimental testing is expected to play an important role in future Darrieus turbine development.

#### References

1. R. E. Akins, "Wind Characteristics at the VAWT Test Facility," Sandia Laboratories Report, SAND78-0760, September 1978.
2. R. E. Akins, "Performance Evaluation of Wind Energy Conversion Systems Using the Method of Bins - Current Status," Sandia Laboratories Report, SAND77-1375, March 1978.
3. M. H. Worstell, "Aerodynamic Performance of the 17 Meter Diameter Darrieus Wind Turbine," Sandia Laboratories Report, SAND78-1737, September 1978.
4. C. W. Dodd, "Lightning Protection for the Vertical Axis Wind Turbine," Sandia Laboratories Report, SAND77-1241, October 1977.
5. R. C. Reuter, M. H. Worstell, "Torque Ripple in a Vertical Axis Wind Turbine," Sandia Laboratories Report, SAND78-0577, April 1978.

#### DISCUSSION

- Q. What are the details of your theoretical prediction? Are you using a steady profile drag coefficient?
- A. Vertical axis technology is not mature, and our analytical techniques are in a state of flux. We presently simulate VAWT aerodynamic performance using a multiple stress tube model which has been tailored slightly to match experimental wind tunnel data. It is not the kind of accuracy we have a lot of confidence in. We are developing additional models.
- Q. What do you have for profile drag in this prediction?

- A. NASA profiles have known values of drag versus angle of attack. We use a multiple stress tube model which uses momentum equations based upon the NACA data and also wind shear assumptions to calculate aerodynamic data for each stream tube element.

The largest errors in the power coefficients have been at high tip speed ratios, which is a relatively lower operation. There is a contract with Jim Strickland at Texas Tech University to develop a two and three-dimensional vortex model. He recently reported that his model corrects the over-predicted performance at high tip speed ratios.

- Q. Did you keep track of the amount of energy used in starting this machine as compared to the total output energy?

- A. I am not able to answer that question precisely, except I know that it's insignificant. Mr. Al says it takes 15 seconds to start.

- Q. Concerning the use of the method of bins, what is the repeatability, and how do you decide which test runs to throw away and which to keep? Also, how do you select the anemometers or the position of the anemometers in relation to cross-correlation?

- A. To my knowledge, no experimental measures have ever been thrown away. They are all averaged equally in making the performance calculations. As for the anemometry, in cases where there is a question about one of the two anemometers shadowing the other, we always use the upstream anemometer.

- Q. Could you comment on the relative differences between using filters and averaging techniques such as that used on the Magdalen Island machine? Have you decided which one does better filtering?

- A. We have never used filtering as I understand you to mean, that is, measuring steady state turbine response. We select a time interval - I think it's typically half a second - and at every time interval we instantaneously measure all of the performance parameters and store them. Each of these parameters is then accumulated according to wind speed. There is no filtering. In other words, there is no compensation for a frequency response of the turbine. This includes having a poor response to a gust direction change, etc.

- Q. The annual COE figures appear surprisingly low in light of the  $C_p$  of 0.41. Would the machine perform substantially better on energy capture if it wasn't run at a constant speed, and how much better could it be?

- A. The energy calculations that we use in all of our studies are based upon 90 percent availability, which is a randomly selected number. They are also based upon the current model of theoretical efficiency, not upon the most recent experimental data to which you refer. Regarding the improvement with variable speed, we feel that a potential energy improvement is in the order of ten percent for a 15 mile per hour median wind speed environment. In our studies we have so far determined that variable speed costs too much and it's difficult to control as well.



TABLE I. - 17 METER TEST TURBINE SPECIFICATIONS

	<u>Original Spec</u>	<u>Present Spec (If Different)</u>
Rotor Diameter	54.9 ft	
Rotor Height	55.8 ft	
Swept Area	2014 ft <sup>2</sup>	
Ground Clearance	16.0 ft	
Overall Height	110 ft	94 ft
Operating Speed	29.8-52.5 rpm	29.8-54.8 rpm
Number of Blades	2 or 3	
Blade Manufacturer	Kaman	Alcoa
Blade Material	Fiberglass/Honey- comb/Aluminum Extrusion	Aluminum Extrusion
Airfoil Section	NACA 0012	NACA 0015
Chord Length	21.0 in	24.0 in
Use of Struts	Yes	No
Blade Length	79 ft	
Blade Shape	Straight-Circular- Straight	
Blade Joints	Pinned	Rigid
Blade Weight (each)	713 lbm	1370 lbm
Strut Weight (per blade)	446 lbm	0 lbm
Speed Increaser	3-Stage Planetary	3-Stage Parallel Shaft
Manufacturer	Crichton	Philadelphia Gear
Speed Increaser Ratio	42.9:1	35.58:1
Belt Drive Ratio to Motor	1.42:1 to 0.8:1	1.7:1 to 0.92:1
Motor/Generator (Induction)	50 hp Squirrel Cage	75 hp Squirrel Cage
(Synchronous)	75 hp	
Brake	Dual Independent 30" Disc	
Brake Torque Capacity	53,000 ft-lb (each)	
Tower OD	20 in	
Tower ID	18 in	
Number of Guy Cables	4	
Cable Angle (to Horizontal)	35°	
Cable Diameter	1 in	
Cable Pretension	12,000-18,000 lb	
Cable Length	129 ft	

TABLE II. - 17 METER TEST CHRONOLOGY

<u>Item</u>	<u>Date</u>
Begin strutted, 2-bladed test	April 1977
Inspect gear box Fit dry sump lubrication Increase motor rating	June and July 1977
Change to strutted, 3-bladed test	December 1977 January 1978
Change to unstrutted 2-bladed test Change gear box	February and March 1979



THE 17-METER TEST TURBINE.

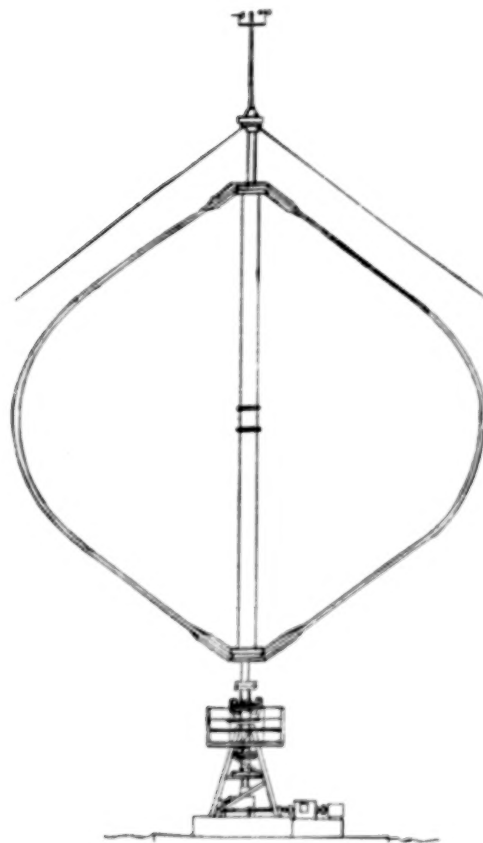


Figure 1.

EFFICIENCY OF THE PLANETARY GEARBOX AFTER DRY SUMP MODIFICATION

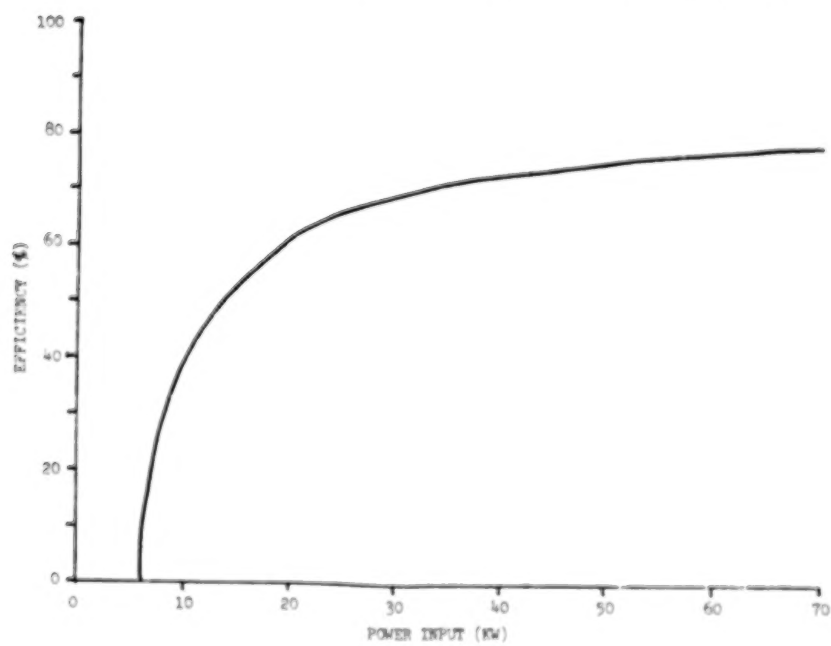


Figure 2.

ROTOR OUTPUT POWER VS WINDSPEED  
 21" CHORD, NACA 0012 BLADE WITH STRUTS  
 52.5 RPM 2 BLADES

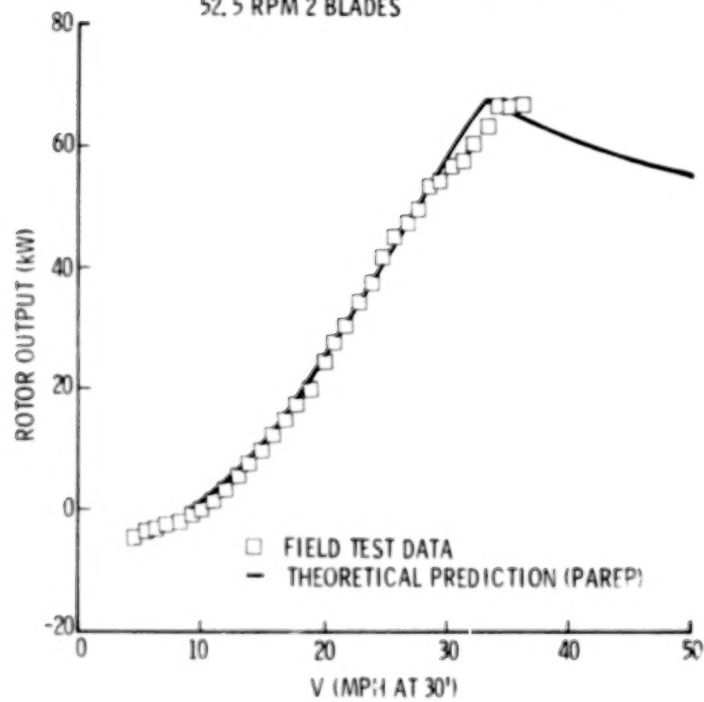


Figure 3.

ROTOR OUTPUT POWER VS WINDSPEED  
 24" CHORD, NACA 0015 BLADE WITHOUT STRUTS  
 50.6 RPM 2 BLADES

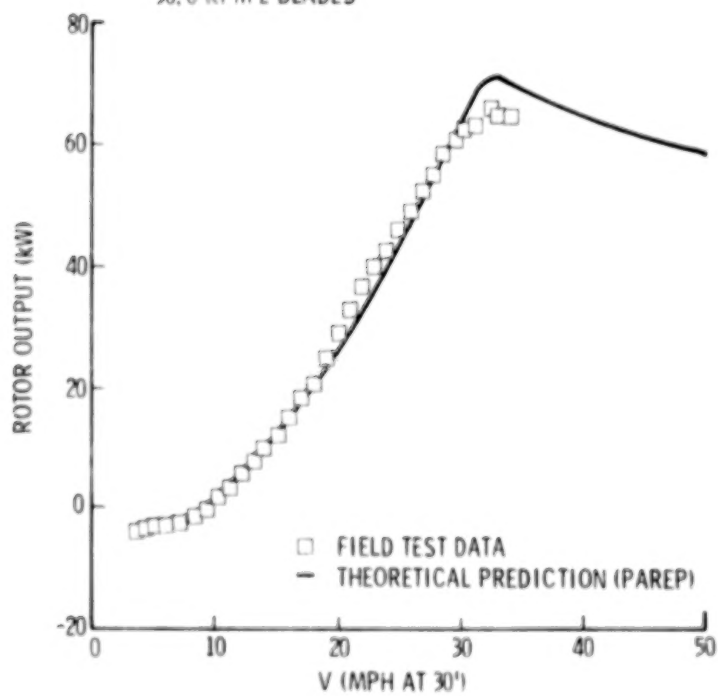


Figure 4.

ROTOR EFFICIENCY VS TIP SPEED RATIO  
 21" CHORD, NACA 0012 BLADE WITH STRUTS  
 52.5 RPM 2 BLADES

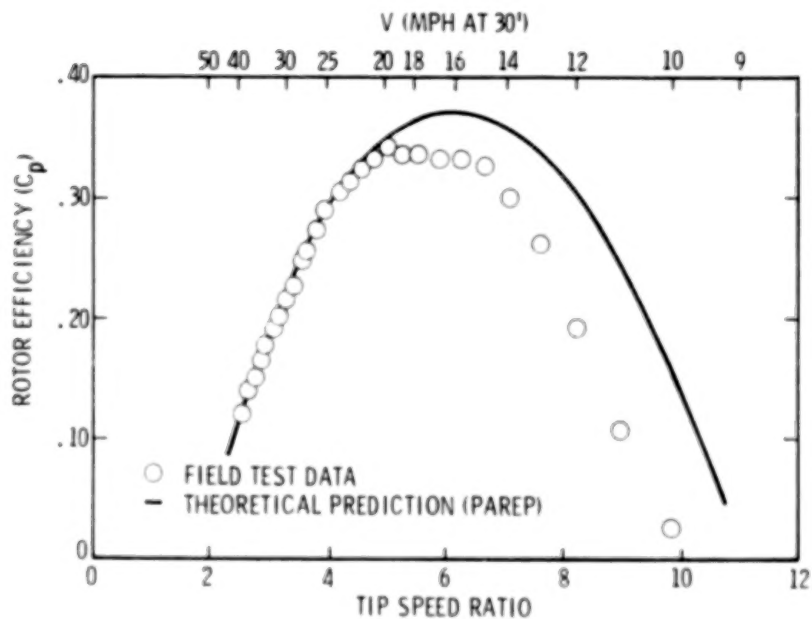


Figure 5.

ROTOR EFFICIENCY VS TIP SPEED RATIO  
 24" CHORD, NACA 0015 BLADE WITHOUT STRUTS  
 46.7 RPM

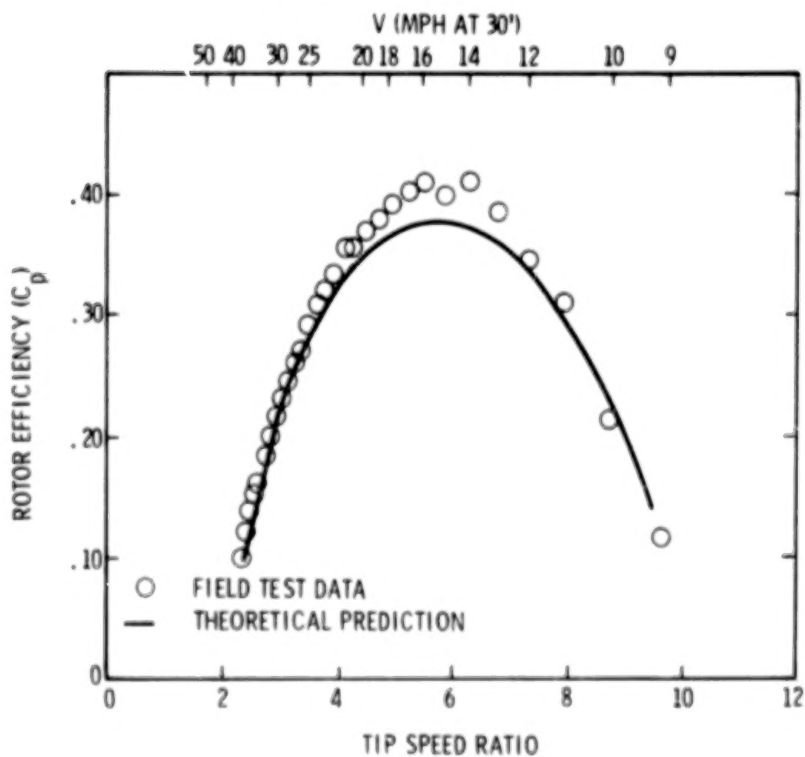


Figure 6.

ROTOR EFFICIENCY VS TIP SPEED RATIO  
24" CHORD, NACA 0015 BLADE WITHOUT STRUTS  
50.6 RPM 2 BLADES

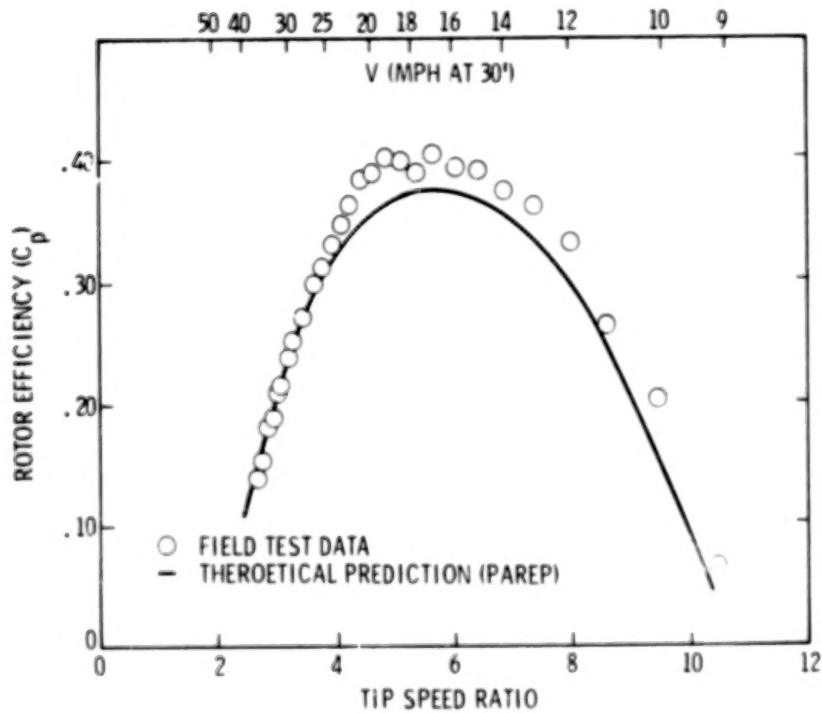


Figure 7.

TWO BLADED ROTOR EFFICIENCY VS TIP SPEED RATIO

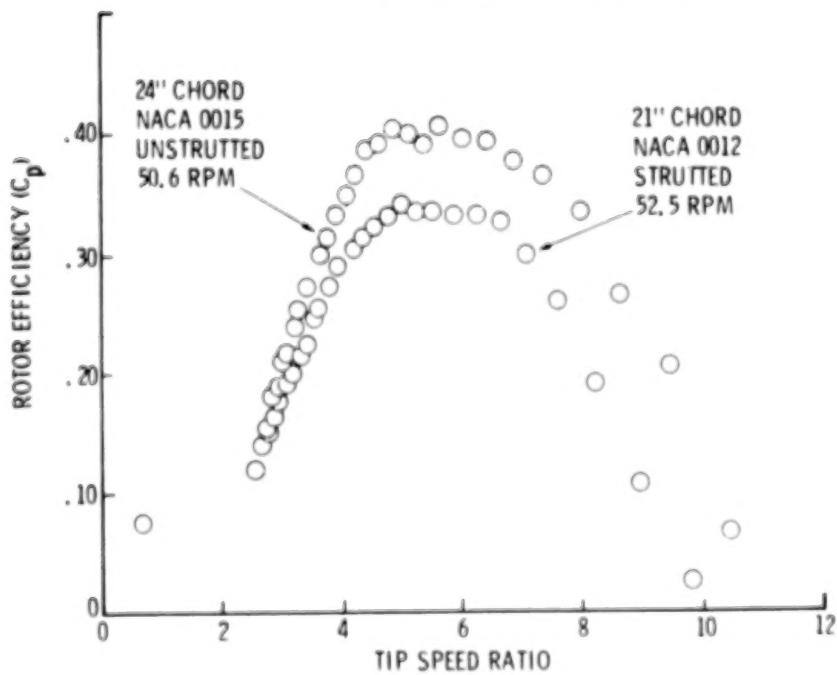


Figure 8.

## PROBLEMS ENCOUNTERED IN 17-METER TESTING

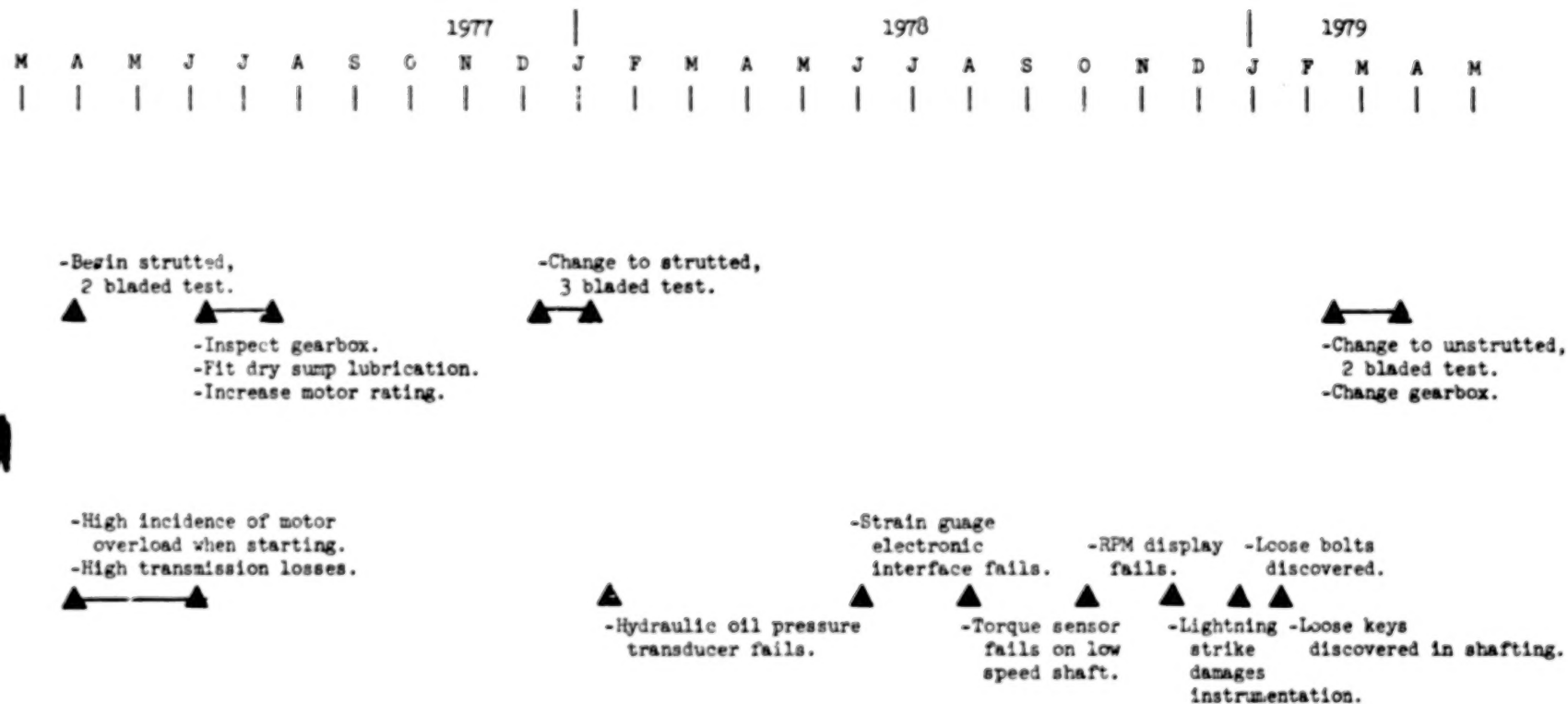


Figure 9

## OVERVIEW OF VERTICAL AXIS WIND TURBINE (VAWT)

### BLADE DESIGN PROCEDURES\*

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The design of a VAWT blade section involves primarily the selection of a manufacturing technology, establishing structural integrity, and obtaining acceptable aerodynamic performance. In this paper, a survey is presented of the practices which have been applied for designing VAWT blades in the past. Through this presentation, an attempt is made to discuss strengths and weaknesses of the existing procedures. Where appropriate, discussion is provided on planned or suggested future work in developing improved design tools.

#### Selection of Manufacturing Technology

This important first step in the design process is governed almost entirely by qualitative issues. Table I lists the features we at Sandia Laboratories have found to be desirable when selecting a manufacturing technology.

It is unlikely that a technology exists which excels in all of these characteristics. Thus, the judgment of the designer is required to make a final selection. Obviously, the relative importance of the items in Table I depends on the particular application. For example, short-term availability may dominate blade selection for a research machine, corrosion resistance for machines destined for coastal use, and so forth. Blade cost, however, should almost always be of primary importance.

In past VAWT blade construction, many manufacturing technologies have been used. These technologies include: aluminum extrusions (hollow and solid), machined aluminum, aluminum extrusion/fiberglass composites, fiberglass/steel, fiberglass, roll-formed and welded steel (straight sections only), advanced composites, and plywood. Of all these, aluminum extrusions have been the most widely used because they possess many of the desirable characteristics of Table I. However, the other listed technologies and promising new proposals (such as composite pultrusions and formed steel blades) should remain as potentially superior candidates to extrusions in certain applications.

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\*This work prepared for the U.S. Department of Energy, DOE, under contract DE-AC04-76DP00789.

### Structural Design

Following selection of a manufacturing technology, critical structural dimensions of the blade section must be determined. At Sandia Laboratories, structural performance is evaluated primarily with numerical (finite element) models. The Canadian National Research Council (NRC)<sup>1\*</sup> has evaluated designs using experimental measurements on scaled wind tunnel models. Both of these methods have been applied on prototype machines which yielded acceptable structural performance.

Considering analytical techniques, blade analysis has focused on static, dynamic, and flutter (aeroelastic) issues. The basic approach has been to design the blade first to static requirements followed by checking and fine tuning (if necessary) to preclude undesirable dynamic or flutter effects.

Static finite element blade models have been developed for quasi-static centrifugal and aerodynamic normal operating loads, gravity loading, and parked-rotor blade collapse in gale-force winds. The MARC non-linear finite element package has been favored for these problems because of significant geometric-non-linearities which occur in the VAWT blade due to the effects of centrifugal stiffening for normal operating loads and large deformations which occur in parked rotor blade collapse.

Table II summarizes the criteria which have been used to determine static acceptability. Typical results for quasi-static blade stresses predicted by MARC are shown in Fig. 1.

The suitability of quasi-static analysis requires system natural frequencies to be well above the load excitation frequencies. Finite element models (using SAP IV primarily) have been constructed to examine resonant frequencies of the complete blade/tower/tiedown system. Typical results from such an analysis, in this case the Sandia 17 meter rotor with two extruded blades, are shown in a fan plot (Fig. 2). Due to the collective effects of conservative static requirements (Table II), the support of the blade at both ends, and the inherent stiffness of the tiedown cable support system, these resonant frequencies are quite high relative to typical excitation frequencies.

This tends to justify the use of quasi-static models. However, efforts are in progress to construct a complete forced-response dynamic model to replace the quasi-static analysis. This is appropriate because economic factors are motivating reduction of conservatism in the static requirements, a trend toward larger rotor height-to-diameter ratios, and

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\*References listed at end of paper.

consideration of alternate blade manufacturing technologies. These changes can lower system resonant frequencies and thereby increase the risk of relying only on static analysis tools.

Aeroelastic flutter instability has been observed<sup>1,2</sup> on VAWT blades. Approximate analyses<sup>3</sup> and experimental data on scale models<sup>1</sup> have indicated that blades meeting the static requirements with section properties similar to aluminum extrusions will have critical flutter speeds well above normal operating speeds. However, there are destabilizing factors which may lower the flutter speeds if alternate sections are considered with substantially different properties than aluminum extrusions. These factors include: the ratio of aerodynamic forces to blade mass and elastic stiffness, the ratio of blade bending to twisting stiffness, and the ratio of blade stiffness to tower torsional stiffness. Efforts are in progress both analytically<sup>4</sup> and experimentally (at Sandia Laboratories) which should yield more quantitative data on the influence of these and other factors on flutter speed.

#### Aerodynamic Design

Aerodynamic design of the blade section is related to the structural suitability of the blade through the shape of the section and the blade chord.

Most Darrieus blades have utilized symmetrical NACA 0012, 0015, or 0018 airfoils, the last two digits representing the percentage ratio of blade thickness to chord. Of these three high lift to drag ratio sections, the 0018 has the advantage (used on the Canadian 200 kW Magdalen Island rotor) of a somewhat higher ratio of flatwise to edgewise stiffness which can improve structural performance. There are insufficient data to clearly distinguish the subtle differences in aerodynamic performance which probably exist between these three airfoils. At Sandia Laboratories, we have favored the 0012 and 0015 airfoils primarily because of a relatively large data and experience base for these airfoils. Undoubtedly, future research should yield a more definitive answer for the most appropriate airfoils, including investigation of series besides the 0012, 0015, and 0018.

A much more significant variable influencing the structural, aerodynamic, and economic performance of a blade is the blade chord, or, more generally, the ratio of blade chord to rotor radius. In general, reducing the chord to radius ratio causes structural section properties of the blade to deteriorate rapidly (see Fig. 3), and the resulting lower rotor solidities also reduce overall aerodynamic performance. These effects tend to drive design toward higher chord to radius ratios. However, blade costs tend to increase as chord to radius ratio increases, which poses a classical trade-off problem for the designer. For extruded aluminum blades on two-bladed rotors, current design practice suggests that the "best" solidity is in the range of 10 to 15%. Based on the trade-offs



involved in this selection, it is apparent that different blade technologies may well yield a different optimum solidity, so the 10 to 15% practice should not be interpreted as a design invariant.

#### Concluding Remarks

The available practices for designing VAWT blades have been applied in designing blades which have provided excellent service on research-oriented machines. However, this is a developing technology and all conceivable phenomena are not included in the analyses. Future efforts at Sandia Laboratories will be directed toward improvement of existing techniques guided by experimental data and operating experience. The output of such an effort can lead to a reduction of technical risks and conservatisms required to cover analysis uncertainties.

#### References

1. R. J. Templin and P. South, "Canadian Wind Energy Program," Proceedings, Vertical Axis Wind Turbine Technology Workshop, SAND76-5586, May 17-20, 1976.
2. B. F. Blackwell, W. N. Sullivan, J. F. Banas, and R. C. Reuter, "Engineering Development Status of the Darrieus Wind Turbine," AIAA J. of Energy, V. 1, No. 1, pp. 50-64, January-February 1977.
3. N. D. Ham, "Aeroelastic Analysis of the Troposkien-Type Wind Turbine," SAND77-0026, April 1977.
4. A. Vollan, "The Aeroelastic Behavior of Large Darrieus Type Wind Energy Converters Derived from the Behavior of a 5.5 m Rotor," Paper C5, 2nd International Symposium on Wind Energy Systems, October 3-6, 1978, Amsterdam, The Netherlands.

Table I. - Qualitative Issues Governing Selection of  
Manufacturing Technologies for VAWT Blades

Economics

- Low Raw Material Costs
- Low Labor Intensity
- Low Tooling Costs

Mechanical Properties of Materials

- Endurance Limits
- Yield Strength
- Density
- Ductility
- Stiffness
- Corrosion Resistance
- Weldability or Joinability

Formability

- Capability to Fabricate High Moment of Inertia, Low Weight Sections  
(1/4 Chord Balance Not Required)
- Capability to Form Curved Blade Sections
- Blade Root Attachment and Shipping Joint Hardware Compatability
- Size Limitations on Chord and/or Blade Length

Availability

- Short-Term R&D Time Requirements
- Long-Term Raw Material Supply
- Energy Intensity for Fabrication and Raw Materials

Table II. - Static Structural Performance Criteria

- Quasi-static vibratory blade stresses less than  $10^8$  cycle endurance limit (approximately 6000 psi for 6063-T6 aluminum extrusions) at normal operating rpm in 60 mph winds.
- Quasi-static blade angle-of-attack changes due to aerodynamic loading less than  $3^\circ$  at normal operating rpm in 60 mph winds.
- Period upwind blade survival for 150 mph static gusts.
- Period gravitational stresses below 40% of yield.
- Blade survival (no yield) at operating rpm + 20% in 80 mph winds (accident conditions).

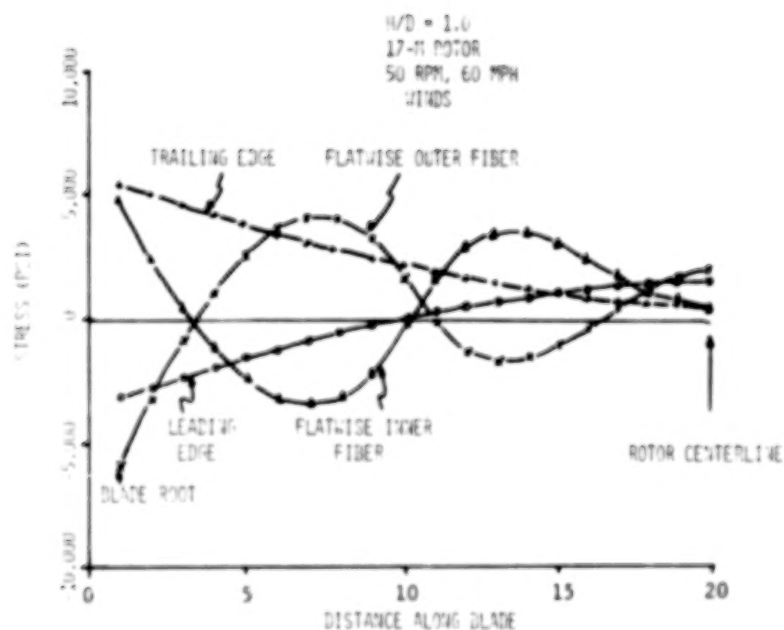


FIGURE 1. Results from MARC Showing Variations in Blade Section Stresses Between the Blade Root and Rotor Centerline. Loading Corresponds to Maximum Aerodynamic and Centrifugal Loads Which Occur in a Revolution at 50 rpm in 60 mph Winds (17-m Rotor).

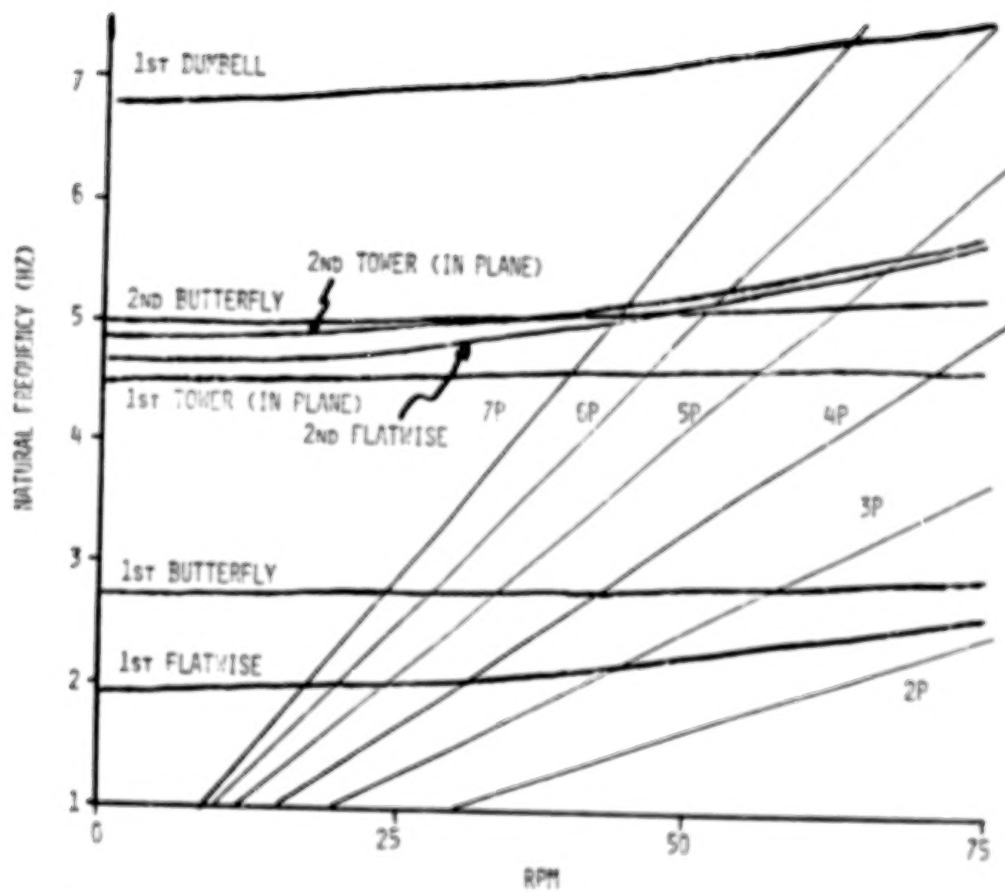


FIGURE 2. System Resonant Frequencies for the Sandia 17 Meter Rotor,

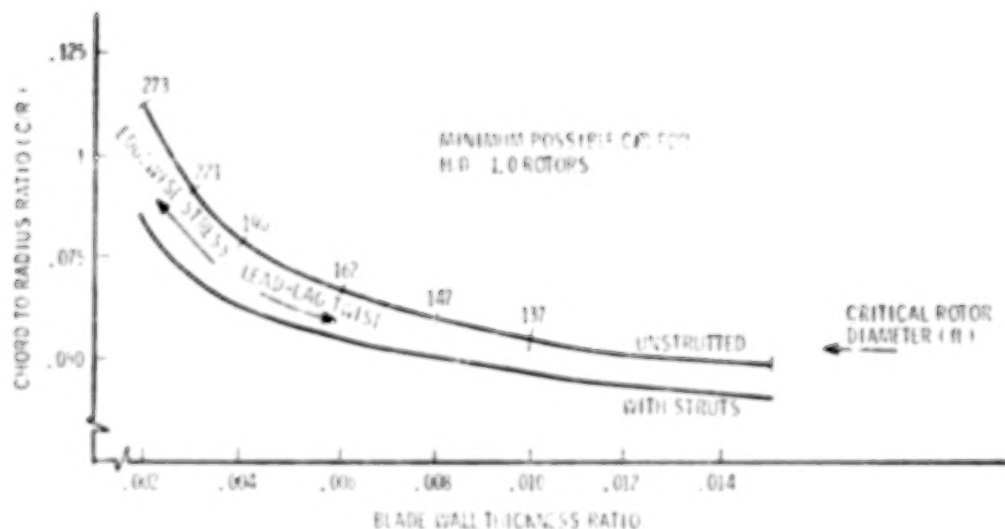


FIGURE 3. The Minimum Possible Chord-to-Radius Ratio for Aluminum Extrusions Satisfying the Criteria of Table II. Blade Wall Thickness Ratio is the Ratio of Wall Thickness to Chord Length. The Effect of Support Struts is Shown. "Critical Rotor Diameter" is the Rotor Diameter Above Which Gravitational Loads Become Excessive.

## FABRICATION OF EXTRUDED VERTICAL AXIS TURBINE BLADES

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An important component of the modern Darrieus type vertical axis wind turbine is the extruded aluminum blade. This is made possible by the requirement that they be hollow, of constant airfoil shaped cross section, and be capable of being bent into a near-troposkein shape about the flatwise axis. They should be light weight, strong, and need a minimum of maintenance. These characteristics describe some important attributes of aluminum alloy extrusions.

Alcoa initiated its wind energy program in 1976 with the fabrication of extrusions for Kaman Aerospace, who furnished the original NACA 0012 53.3 cm (21 in.) chord blades for Sandia Laboratories' 17 m research VAWT.<sup>(1)</sup> A typical section of these blades (Fig.1) contains a D-shaped nose extrusion and vee-shaped trailing edge extrusion. Honeycomb provides the filler for the balance of the blade section, and the assembly is covered with glass fiber reinforced plastic.

Alcoa Laboratories also furnished formed hollow extruded 15.3 cm (6.03 in.) chord blades for the Sandia 5 m research machine shown in Fig. 2. The blades were extruded as 6061-T4, bent to the straight-curve-straight approximate troposkein contour, and age hardened to -T6 temper.<sup>(2)</sup> Several other sets of these blades have since been furnished for other experimental and commercial VAWT projects.<sup>(3)</sup>

A unique project involving a 10 kw Darrieus VAWT is now in operation at Clarkson College in Potsdam, New York. (Fig. 3) The Mechanical Engineering Department at Clarkson sponsored a program with contributions from Agway, Alcoa, Allen-Bradley Company, Chromalloy Farm Systems, Niagara Mohawk Power Company, Reliance Electric Company, Sign-X Laboratory, PCB Piezotronics Corporation, and Unarco-Rohn Tower Division, in which the VAWT is mounted on a typical silo. The silo provides a solid base and extra elevation to take advantage of the increased wind power at moderate elevations. The special cantilever mounting system places the rotor above the top of the silo, where there are few, if any obstructions to the wind, and no guy wires are used which would obstruct normal farming operations.

In late 1978, Alcoa Laboratories furnished a set of 61 cm (24 in.) NACA 0015 blades to Sandia for the 17 m research turbine. These blades were extruded in 24.5 m (80 ft.) lengths in alloy 6063-T6, and formed by Alcoa personnel at Sandia. The forming press was shipped on a low-boy trailer and set up in a vacant hangar where the forming was completed. An overall view of the forming setup is shown in Fig. 4. The 61 cm blade is nearly complete in this photograph.

Alcoa's Lafayette, Indiana extrusion facility is one of the largest in the world, containing a number of presses and associated equipment, in-

cluding two extrusion presses of 14,000 ton capacity. Utilizing this large equipment, we have demonstrated the capability to produce 74 cm (29 in.) chord blades, large enough to equip a turbine with rated power over 500 kw. Fig. 5 shows the 74 cm blade beside the 15 cm blade for comparison.

An intermediate blade size with a 35.5 cm (14 in.) chord width is available for VAWTs in the 30 to 90 kw capacity range.

The design breakthrough came in 1976, when experimental data was made available that showed that mass balance at the quarter-chord point was not necessary for Darrieus vertical axis turbine blades. (2) (4) This discovery permitted design of cost-effective extrudable blade shapes. Some of the basic principles of the extrusion process may help explain how this is accomplished.

The extrusion process is basically hydraulic--causing hot plastic metal to flow through a die under pressure. Extruded shapes fall into three classes--solid, semi-hollow, and hollow. Solid shapes have no internal voids such as bars or angles; semi-hollows are solid shapes that have deep channels like window framing members; hollow shapes have totally enclosed voids, for example, tubes or our blade extrusions.

We have different size presses for a variety of reasons. Harder alloys require more pressure, larger sizes of shapes require large dies and presses to handle them, and some shapes are just harder to extrude than others and require more pressure.

In Fig. 6, the basic solid shape die set is shown, consisting of die, die holder, die block and tool carrier. All this fits into the press cylinder. The hot metal billet is pressed against the die and under sufficient pressure flows through the tee-shaped opening and exits down the runout table as a structural tee.

To make hollow shapes, the die is either a bridge or porthole die. (Fig. 7) The bridge or front die is used to support a mandrel at the center of the external opening, and the metal flows between the mandrel and the die to form the inside and outside contour of the hollow. Only certain alloys like 6061 or 6063 can be used in this process because the billet is actually split into segments in the front die and welded back together inside the die under the heat and pressure of extrusion. Other alloys do not attain sound welds under these conditions. The desired shape emerges as a one-piece hollow extrusion, ready for heat treatment and final straightening. Alloy 6063-T6 is a moderate strength, ductile material and can be fabricated in many ways. The VAWT blade extrusion is normally shipped in approximate 12 m (40 ft.) lengths, although longer lengths up to 26 m (85 ft.) are available on special inquiry. These are bent to the contour of the approximate troposkein using a large hydraulic three-point bender. After the desired contour is reached, they are cut to exact length and prepared for joining or terminating.

A typical blade design is that evolved from the low-cost 17 meter VAWT we are doing for DOE.<sup>(5)</sup> The turbine operating and loading requirements resulted in the design of the 61 cm chord blade. (Fig. 8) It is a 6063-T6 hollow extrusion with a weight of approximately 27 kg/m (18 lbs/ft). Economic considerations of size, truck trailer length, handling and fabricating suggested limiting the piece length to about 12 m (40 ft.).

Blade end-to-end splices (Fig. 9) will be required when the total length between points of attachment is longer than the shipping length of 12 m (40 ft.). Stress analysis of the blade at maximum crisis load (75 rpm, 35 m/s) and buckling under maximum survival wind gust load (67 m/s) showed that the joints can often be safely located in a minimum stress zone. The lengths of the blade sections will be finally adjusted to obtain this position for the joints. These stresses are approximately 22 Mpa (3200 psi) in tension due to centrifugal forces, combined with  $\pm$  12.4 Mpa (1800 psi) in bending due to aerodynamic loads. The bending loads are cyclic, depending upon whether the blade is at a position of maximum or minimum lift. Insert sections are sized and proportioned to satisfy fatigue strength criteria and a flatwise bending stiffness across the joint. The extruded inserts of 6061-T6 alloy have been designed to fit closely into each of the hollows in the blade. Counter sunk head blind rivets will be used to attach the inserts to the blade, and a layer of epoxy will be applied before riveting in place to prevent movement which can lead to fretting fatigue failures. When finished, the joint will present an airplane wing appearance, have minimum aerodynamic resistance, and fulfill structural design criteria of restoring full tensile strength across the joint.

The solid blade-to-torque tube connection (Fig. 10) is the result of several design iterations prior to selection of the final concept. These have included socketed type connections which employed metal castings that were welded and/or bonded to the ends of the extrusion, flexible connectors which allowed the blade to float, epoxy bonded systems which were safety clamped, and mechanical clamping techniques that enabled the blade to be rigidly attached to the central torque tube mounted to the central tower.

Analyzing the forces, it became apparent that the loads were not difficult to resist. However, it was estimated that the connection may undergo one billion stress cycles during the 30-year life of the turbine. For this type of life cycle criteria, it was necessary to reduce component fatigue stresses to an absolute minimum.

This fitting was required to be rigidly connected to hardware which attached to the torque tube. Concepts employing cast or welded members for this connection became very large and expensive and were ultimately abandoned. As the concept evolved, the use of a simple end fitting on the rotating torque tube, together with a trusslike stiffener, allowed the connector parts to be simple to fabricate, easy to procure from common stock, and relatively light in weight. Additionally, the aerodynamic drag of this rotating mass was reduced by opening this attachment area to allow air to pass through. This stiffener can then be welded to one



of the torque tube flanges in the factory or sandwiched between the mating flange surfaces and secured to the flange using the flange through bolts in the field.

The blade assembly is bolted to the bracket and the stiffener. Safety devices on the fasteners are used to prevent them from loosening during service.

The most unique element of this blade connector is an extruded aluminum blade attachment fitting comprised of two extrusions, approximately 2 m (6 ft.) long, which are welded to the ends of the straight blade sections. The welded attachment was first conceived to grip the blade at the extreme leading and trailing edges and transfer the loads to the end fitting. Upon closer study, concern developed that the extreme fiber tensile stress in the trailing edge would not be capable of withstanding the estimated billion-cycle fatigue criteria. Moving the attachment point inward linearly reduced the weld joint fatigue stress, thereby improving reliability for 30 year life without blade replacement. Care should be taken to assure that this blade assembly weld is surface ground and peened to reduce any stress risers. This surface was tapered to ensure a gradual stress transition. This design is considered to be a cost-efficient, sound approach to a complex problem, and will be used for the four demonstration machines expected to be constructed on this contract.

Marketing studies (6) have already given us information about future generations of large wind energy conversion systems, and the emphasis of this workshop is on these large machines. It is of interest, therefore, to consider what role extruded blades can play in the future for large vertical axis machines. (Fig. 11)

The first composite blades by Kaman Aerospace showed the feasibility of using extrusions with other high strength structural materials. It is reasonable to expect that airfoils of about 1.5 m chord could be fabricated with one or more extrusions and honeycomb intermediate structures.

We are also looking at extrusion designs to be used together and with heavy sheet or plate components (Fig. 12) Assembly tolerances and joining practices will require significant investigation, because typical assembly dimensions would be in the range of 23 cm across the blade, and standard extrusion tolerances would be  $\pm 0.16$  cm. Further, twist and bow tolerances need to be considered. These shapes are long and flexible, so standard twist tolerances of 3 to 5 degrees should be satisfactory. Bow is the longitudinal deviation from a straight line and is more critical for long assemblies. The industry standard bow tolerance in a 12 m length is 1.0 cm, so jigs and fixtures need to be designed to pull in at least this much.

Consideration of blade-to-blade joints and blade-to-tube terminations will be needed. A concept that has been proposed but not completely designed would look like a bolted pipe flange, except, of course, airfoil shaped, which would be welded to the blade ends and mated in the field in a straight-forward manner. The largest area of question in this design is the design of

the welded joint at the leading and trailing edge where blade stresses have been calculated to reach 55 Mpa (8000 psi) to 62 Mpa (9000 psi). Blade-to-torque tube assemblies using the reinforced stiffener should be relatively easy to make, based upon our current experience.

These ideas are still being improved upon and by no means are the only ones to be considered. We have attempted in every case to consider field application problems and avoid the very sophisticated where factory-controlled environment is needed. Consultation with erection engineers and other building construction disciplines has been helpful in pointing out pitfalls before they become designed in, and costly to accomplish.

Alcoa supports development of our wind energy resource in the the most practical way possible, and hopes that these blade design efforts will help provide a cost-effective component in its development.

#### References

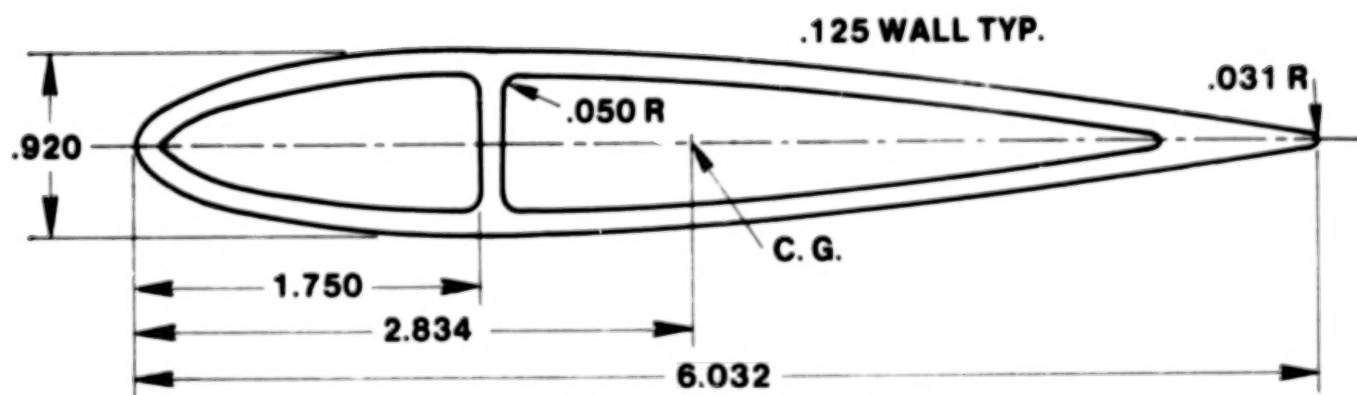
- (1) Borada, J. J., "Blade Design and Fabrication of 17-m VAWT" Vertical-Axis Wind Turbine Technology Workshop, Pgs. II - 205-221, May, 1976.
- (2) Ai, D.K., "A Low Cost Blade Design for a Darrieus Type Vertical-Axis Turbine", Wind Technology Journal, Vol. 2, Spring-Summer, 1978.
- (3) Rollins, J.P., Kear, E.B., Jr., "Progress Report on Silo-Supported Darrieus Wind Turbine" presented at Annual Meeting American Wind Energy Association, Hyannis, Massachusetts, September, 1978 (To Be Published).
- (4) Ham, N.D., "Aeroelastic Analysis of the Troposkein-Type Wind Turbine" Vertical Axis Wind Turbine Workshop, Pgs. II - 185-204, May, 1976.
- (5) Ai, D.K., et al, "Design and Fabrication of a Low Cost Darrieus Vertical-Axis Wind Turbine System, Phase One-Technical Report", U.S. Department of Energy Cont. EM-78-C-04-4272, 1979 (To Be Released).
- (6) Garate, J. A., "Wind Energy Mission Analysis" Proceedings, Third Wind Energy Workshop, Vol. 1, p. 209-218, J.B.F. Scientific Corp., 1977.

### Discussion

- Q. Has the effect of residual stresses from bending on the overall strength of the blade been investigated?
- A. This problem has been recognized and evaluated in a qualitative sense. The bending operation is confined to the outer periphery which is a relatively low stress area where the combined stresses would not be expected to approach the endurance limit. During bending, we use shaped dies to avoid developing kinks or sharp corners which would create stress concentrations.
- Q. The reason for the previous question concerns the effect of surface defects in the extrusion causing increased stresses. What is the type of surface finish in the as-extruded condition?
- A. Quality control procedures during ingot production, die construction, extrusion and final inspection are constantly on the alert to avoid surface defects such as die lines or inclusions. These are nearly always detected and corrected or replaced at the plant. Surface finish standards call for 100-150 RMS finish on the as-extruded exterior surface, with no individual defect exceeding .002".



Figure 1. Kaman blade cross-section.



# COORDINATES

X	Y	X	Y
.000	.000	1.838	.460
.020	.076	2.450	.444
.077	.145	3.063	.405
.153	.200	3.675	.349
.306	.272	4.288	.281
.459	.322	4.900	.201
.613	.359	5.513	.111
.919	.409	5.819	.062
1.225	.439	6.005	.031
1.531	.455		

Figure 2. - Alcoa 6 inch (0.15 m) all aluminum blade (all dimensions in in.).

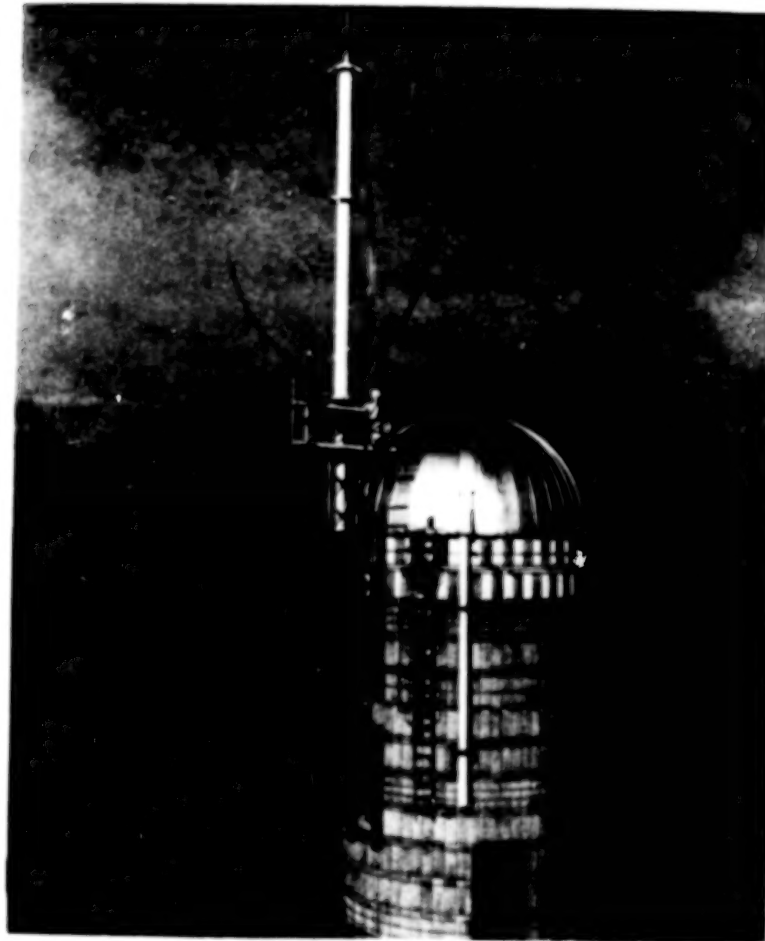


Figure 3. Clarkson College silo-mounted VAWT.

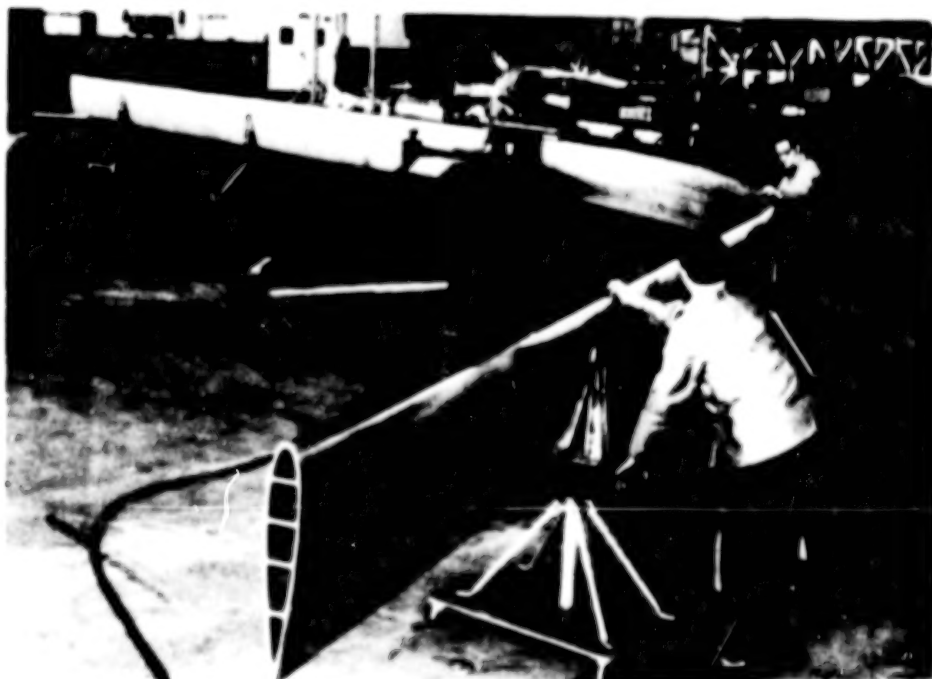


Figure 4. Alcoa 61 cm (24 in) blades formed at Sandia Laboratories.

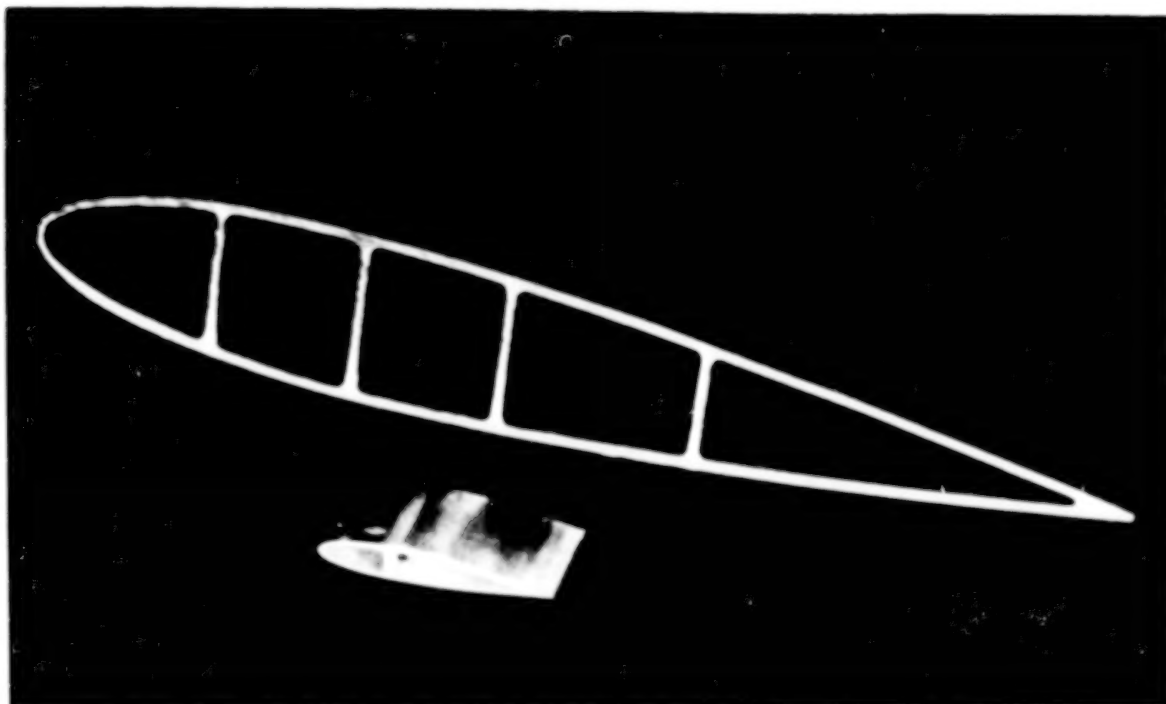


Figure 5. Alcoa 74 cm (29 in.) and 15 cm (6.03 in.) blades.

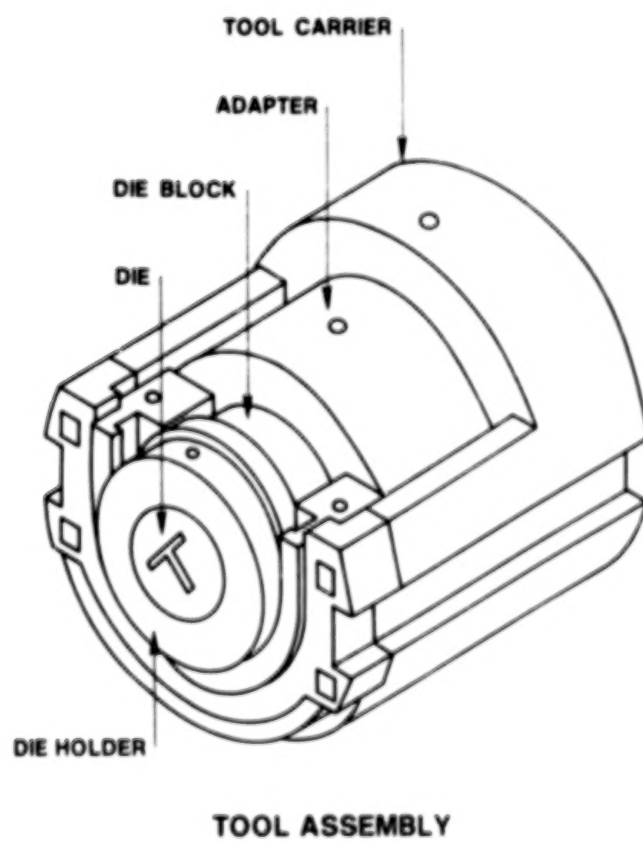


Figure 6. Typical solid die set

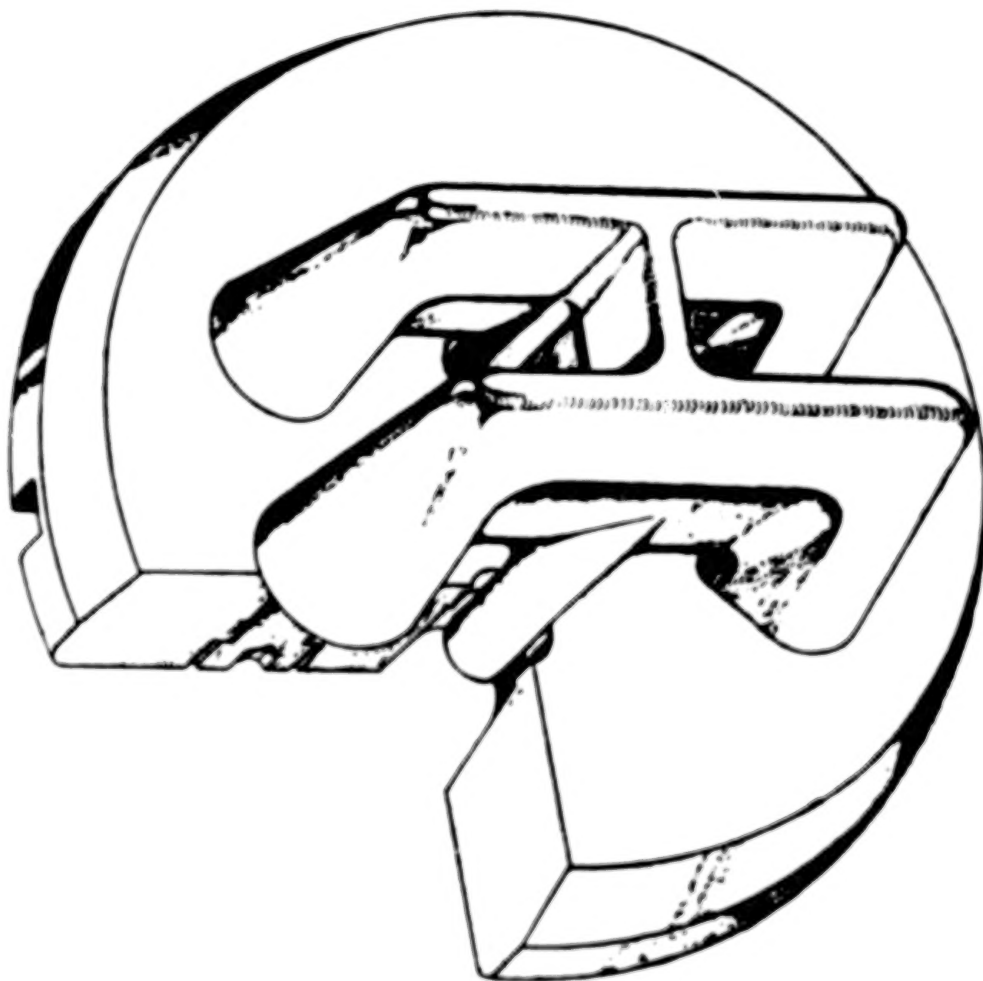
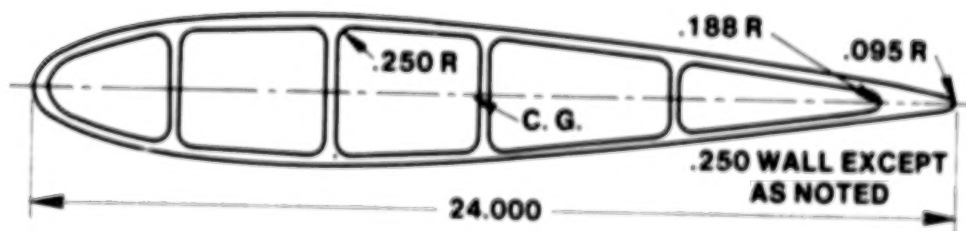


Figure 7. Typical bridge die assembly.



### BLADE CROSS SECTION

Figure 8. Alcoa 61 cm (24 in.) blade section.

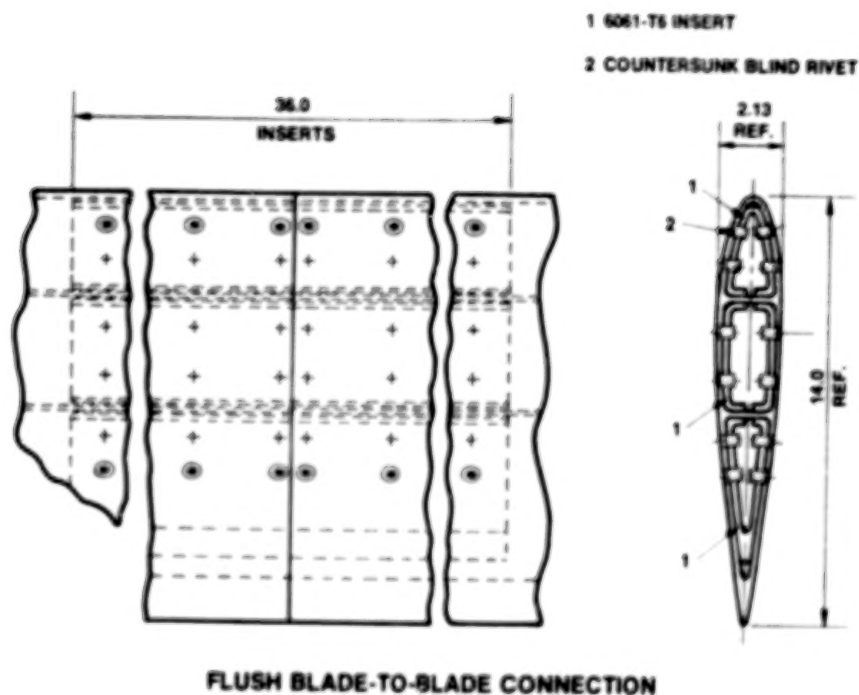


Figure 9. Typical end-to-end blade splice.

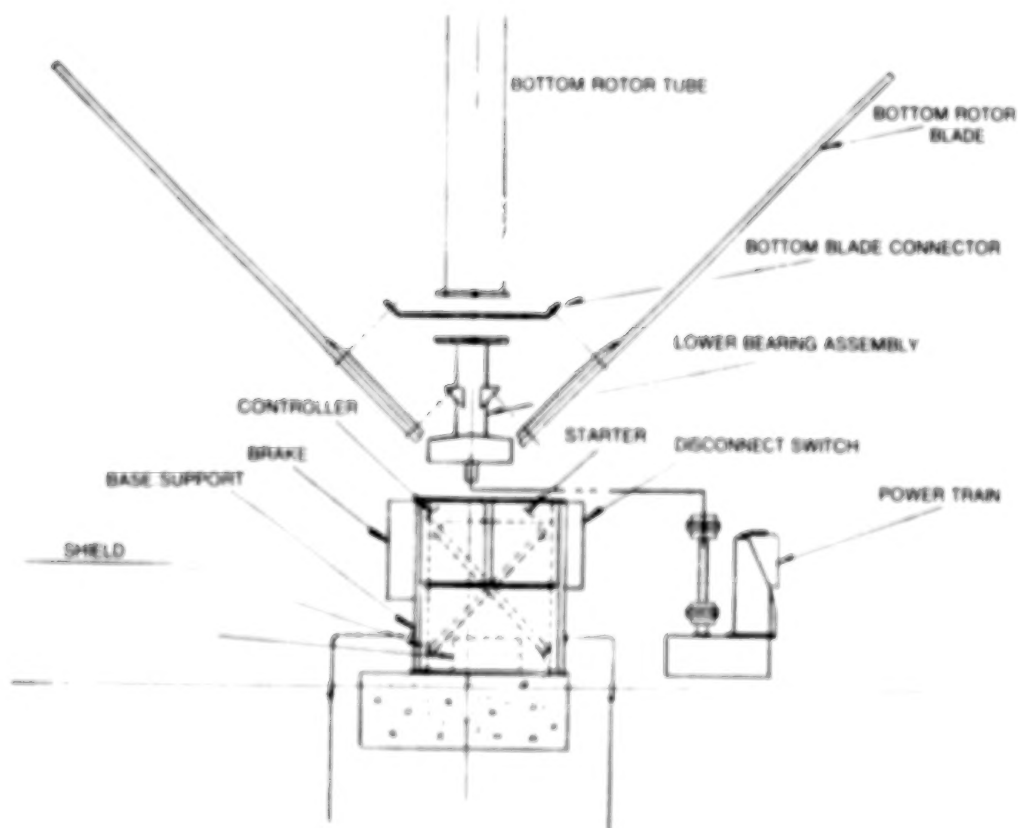


Figure 10. Reinforced blade/torque tube connection



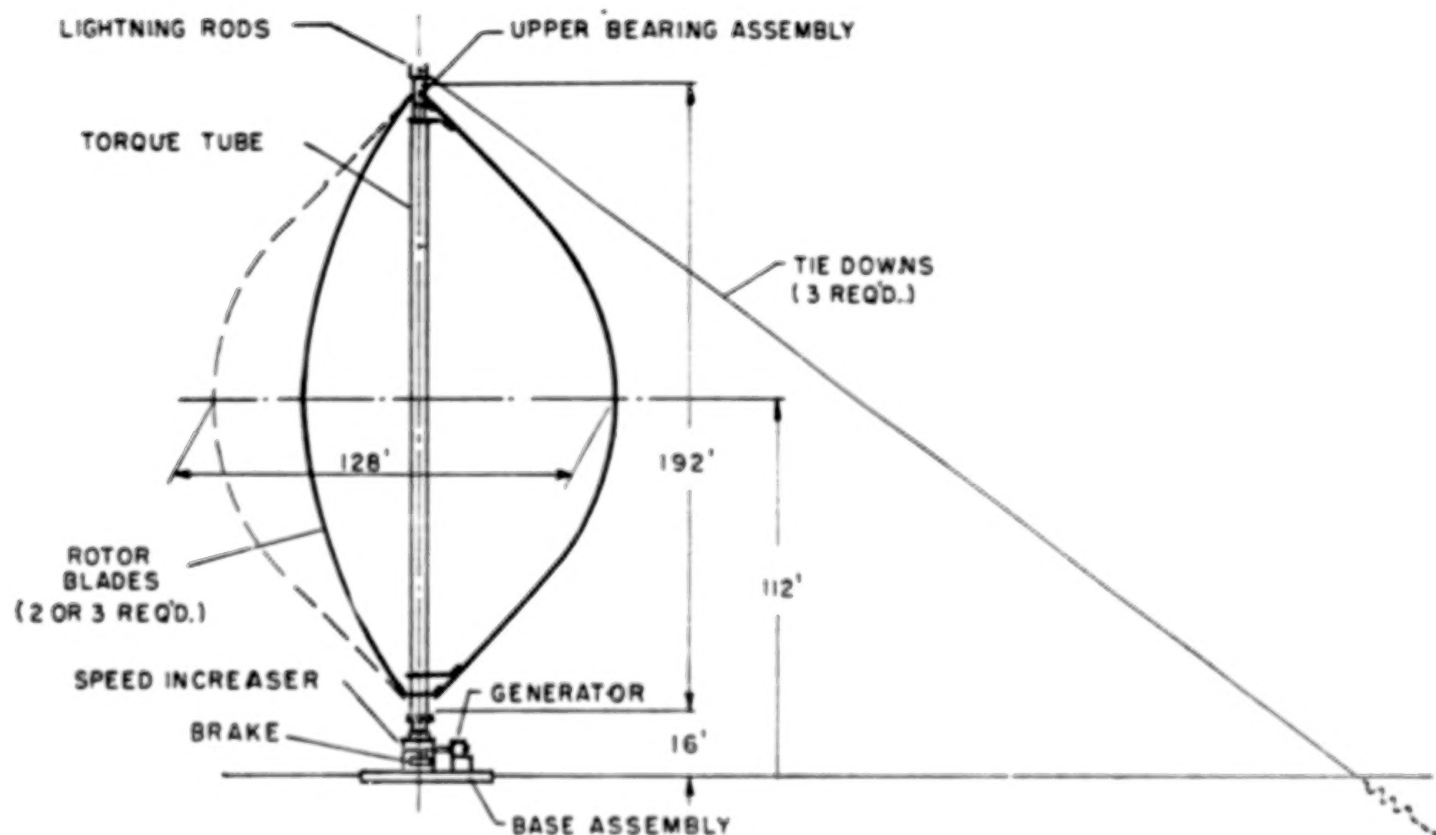


Figure 11. - Nominal 1 MW ALVAWT. Alcoa vertical axis wind turbine.

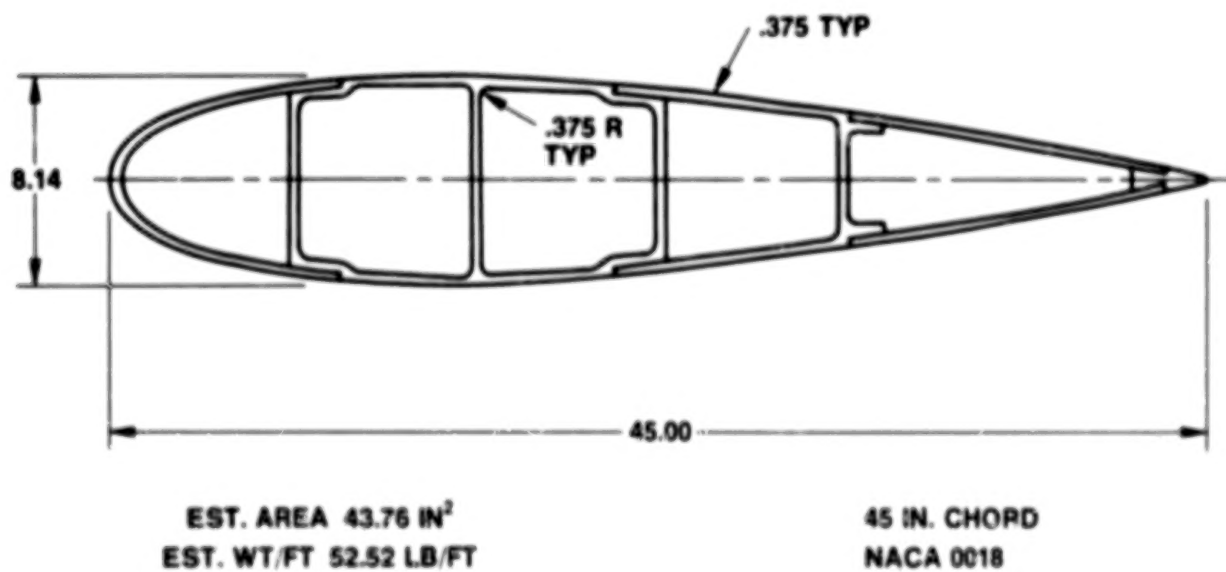


Figure 12. Alcoa 115 cm (45 in.) chord blade.

## OPERATIONAL EXPERIENCE WITH VAWT BLADES

### AT SANDIA LABORATORIES\*

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Sandia Laboratories has operated three Darrieus turbines (2 meter, 5 meter, and 17 meter diameter rotors) at its test facility for the last several years. Through this test program, a variety of blade types and rotor configurations have been tested for structural and aerodynamic performance. This paper will discuss primarily blade structural performance aspects of the tests on the 17 meter rotor.

The first blade installed on the 17 meter rotor was fabricated by Kaman Aerospace Corporation. The Kaman blade, shown in Fig. 1, is a helicopter-type composite of aluminum and fiberglass. The airfoil is a NACA 0012 with a 21 inch chord. A single blade is made up of five individual sections (two straight sections, two struts, and one curved section) joined by flatwise-free pins (Fig. 2). The blades are instrumented with direct strain gages bonded to the extruded aluminum spars at the locations shown in Fig. 2. The rotor was initially configured with two blades and after about 8 months of testing, a third blade was added.

The performance of the Kaman blade was quite acceptable. No maintenance was required and no blade deterioration was evident upon removal of the blades. Installation was tedious because of the many individual blade sections and difficulties in aligning the pin connectors. A high frequency (above 4/rev) blade resonance in the lead/lag strain gages was observed in the three-bladed configuration at one test rpm (45.5). This resonance was substantial only with winds above 35 mph and was almost undetectable at rotor rpm's 5% on either side of 45.5. Normal operating rpm for the 17 meter rotor is about 50 rpm. No similar resonances were observed in the two-bladed rotor, apparently because of the lower excitation frequencies present with two blades.

Test results for typical steady and vibratory stress measurements are summarized in Figs. 3 to 5. The details of the data reduction and measurement techniques are discussed in an earlier report.<sup>1</sup> Also shown on Figs. 3 to 5 are predicted values of the stresses based on the MARC quasi-static finite element model. In general, the agreement is very good although scatter in the vibratory data is substantial due to difficulties in measuring the windspeed actually experienced by the blade. The data for the edgewise strain (Fig. 5) do exceed predictions somewhat for winds above 40 mph. We believe this is due to a dynamic excitation of the first lead/lag blade mode (the "butterfly" mode)

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\*This work prepared for the U.S. Department of Energy, DOE, under contract DE-AC04-76DP00789.

<sup>1</sup>W. N. Sullivan, "Preliminary Blade Strain Gage Data on the Sandia 17 Meter Vertical Axis Wind Turbine," SAND77-1176, December 1977.

by 3/rev edgewise force frequencies. A crossing of the 3/rev line with this rotor frequency is predicted at 55 rpm, which is quite near the 52.5 operating rpm shown.

Future testing efforts are planned to center around continued examination of existing data to identify important structural phenomena and guide the development of analysis models. Additional test series are also being planned to expand the data base. These new test series include modal analysis tests on the 17 meter rotor to experimentally determine frequencies and mode shapes, dynamic strain measurements on the 17 meter rotor with the new Alcoa extruded blades without support struts, and flutter tests on the 2 meter rotor.

#### DISCUSSION

Q. With regard to the very high wind load cases, can one consider having to orient the rotor to a less vulnerable position in high winds?

A. Yes. For two-bladed systems, the rotor is substantially less vulnerable to buckling with the blade chordline oriented parallel to the wind velocity. Mechanical systems to provide this orientation for the rotor may be worthwhile on larger systems where such mechanisms may cost a fairly small fraction of the total.

Of the design requirements that we try to use as guidelines, there is not a single one which totally dominates the design. Thus, if we by some means eliminate the buckling problem, either by lowering the design wind speed or developing an attenuating mechanism, the edgewise blade stress, for example, still prevents us from significantly reducing blade section properties. In his paper, Mr. Kadlec talked about changing all of the design requirements. That is what is required to save blade weight on future designs.

Q. Has anyone considered introducing a lead-lag pin in order to get rid of these edgewise stresses, reduce the butterfly mode, and get some attenuation on torque ripple?

A. I think that the Magdalen Island machine, through the linkages on the struts, has a form of lead-lag hinging, or at least a lead-lag damping. We have not considered too actively the lead-lag hinge because, I guess, the hub diameter is relatively small. With small diameter hubs, the blade must lead the tower quite a bit to get the torque out of the blade. I do think that kind of activity may well be appropriate in the future, just as the teetered hub evolved for horizontal axis systems. We still are trying to keep these new ideas in mind as we proceed.

- Q. In the last two slides that were presented, it appeared that the results and experimental data were on a different slope in both cases, and there appeared to be a significant variation in the magnitude of the results. Do you feel that the source of the variation is atmospheric in nature - turbulence, or relating to something else? Also, do you feel the MARC Program is predicting the proper phenomena that you are measuring, and what is causing the variation, in your opinion?
- A. I think you may have alluded to the scatter in the data. It is just tremendously difficult to measure blade strains and wind speed at the same time. I believe that the anemometer reading is not necessarily reflective of the wind speed that occurs all over the disc at the time that it was rotating, and I think that induces tremendous scatter. The slope of the data and theory is in reasonable agreement in the flatwise direction. There is about as much data below the prediction line as above it. In the case of the edge-wise stresses, I agree there is a different slope, and I believe it is due to dynamic effects that are inherently neglected in the model. We happen to know that we are relatively near a resonance frequency, and I think that is what is causing it.
- Q. You use the static criteria for a machine that is designed for 30 years. How confident are you in using the static criteria after it was subjected to dynamic loading? Secondly, what kind of dynamic loading has been included?
- A. I think the dynamic factors are important enough to warrant being quite conservative in the static design requirements. I think you are asking me whether I think the static design requirements are conservative enough. The answer is I think they are, but only because I have seen the data on machines that have been designed to those requirements. I will not deny that there is an element of judgment involved, and that the risk of some disastrous dynamic effects remains. This is why we plan to expand our efforts to analyze and include dynamic effects throughout the design process.

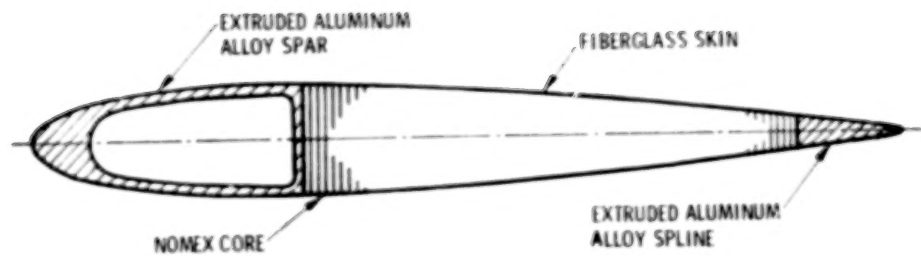


FIGURE 1. Cross Section of the Kaman Blade. Blade Chord is 21".

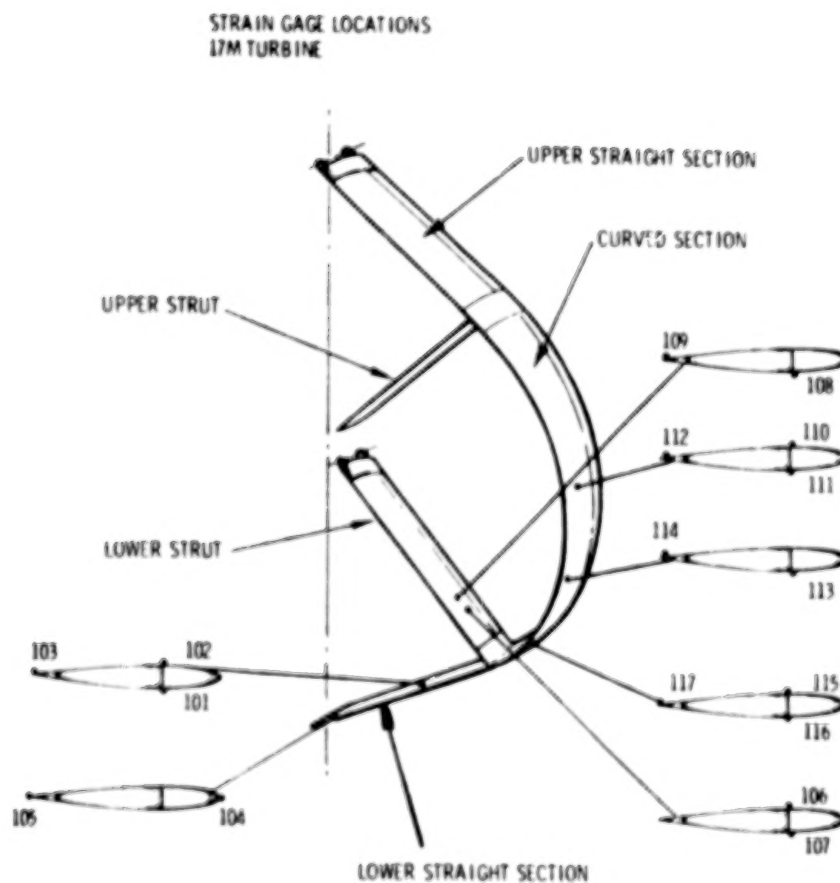


FIGURE 2. Kaman Blade Geometry Indicating Locations of the Strain Gages.

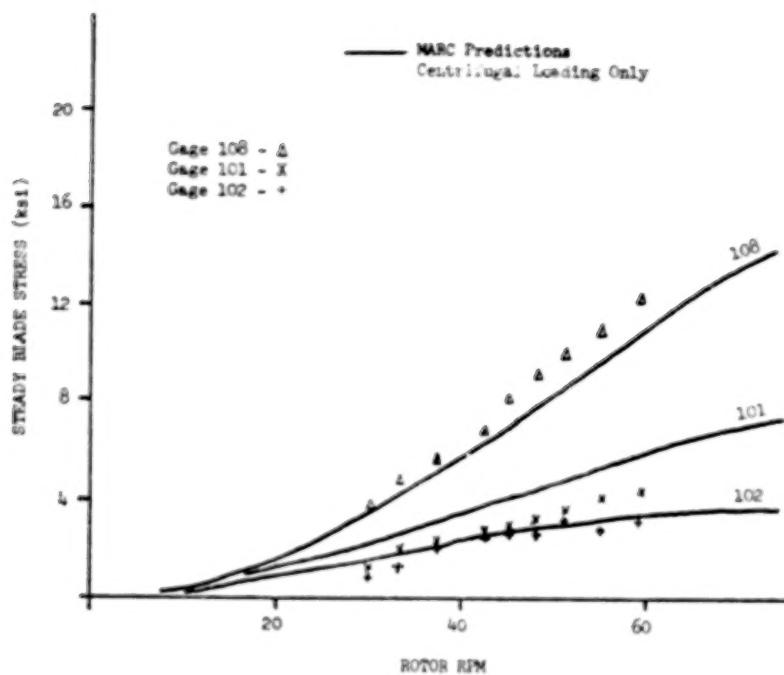


FIGURE 3. Steady Stresses as Measured and Predicted for the 17-m Rotor. Steady Stresses are Measured by Operating the Rotor in Negligible Wind.

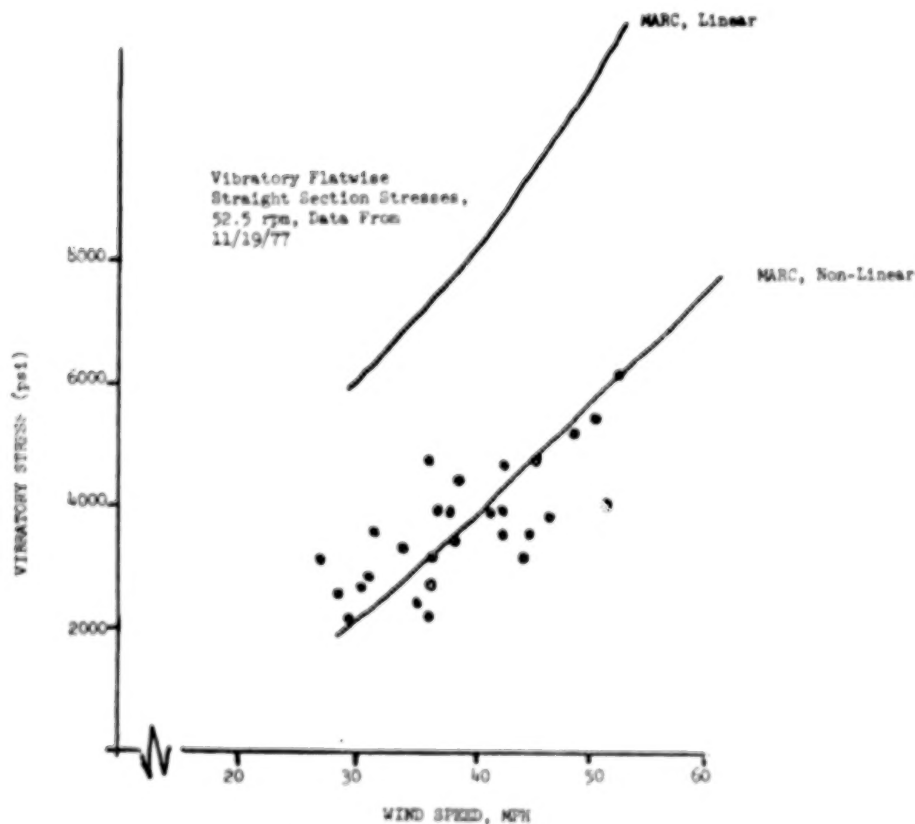


FIGURE 4. Flatwise Straight Section Vibratory Stresses at 52.5 rpm as a Function of Windspeed. MARC Quasi-Static Analysis Also Shown, With and Without Non-Linear Options.

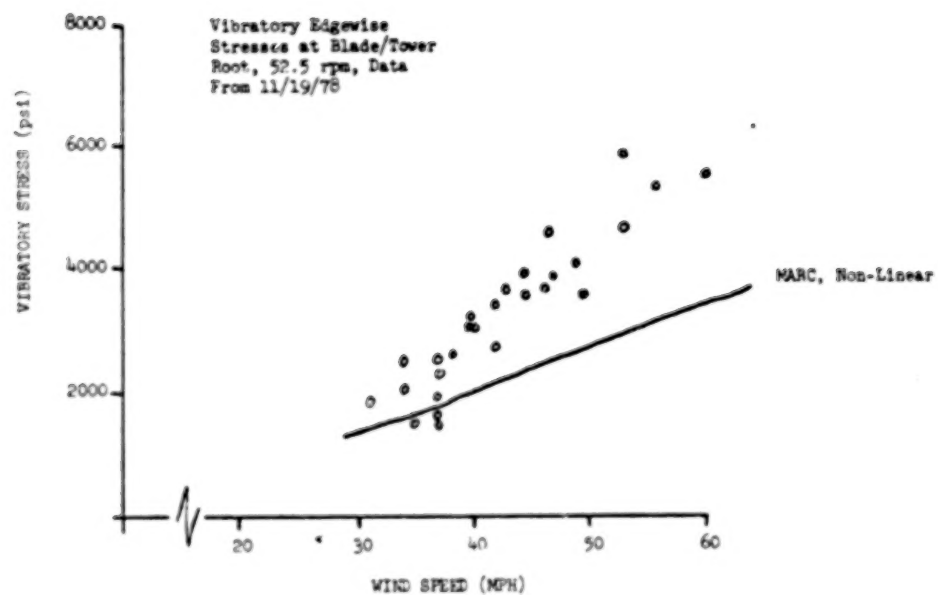


FIGURE 5. Edgewise Straight Section Vibratory Stresses at the Blade/Tower Attachment Region.

## STRUCTURAL ANALYSIS CONSIDERATIONS FOR WIND TURBINE BLADES

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### SUMMARY

Items which should be considered in the structural analysis of a proposed blade design are briefly reviewed. These items include the specifications, materials data, and the analysis of vibrations, loads, stresses, and failure modes. The review is limited to the general nature of the approaches used and results achieved.

### INTRODUCTION

Wind turbine blades are being designed in a variety of configurations and are being manufactured from a variety of materials. It is the task of the structural analyst to verify that a particular design satisfies all requirements concerning structural integrity. These requirements include freedom from failure mechanisms such as fatigue, buckling, yielding and fracture, and limitations on deflection, wear, and corrosion. The purpose of this paper is to briefly review the items which should be considered when planning the structural analysis of a wind turbine blade. These items include specifications, materials data, and the analysis of vibrations, loads, stresses, and failure modes. Specialized methods for performing these analyses will not be discussed in this review, but rather the general nature of the approach and the results.

Among the many critical components in a wind turbine system, the blades are usually considered to be the most difficult to design. Typically, wind turbine blades are long flexible rotating airfoils which continuously sustain cyclic and transient loads. They must operate efficiently at off-design conditions because of the variability of wind speed and direction. As flexible airfoils, they are subject to aeroelastic and mechanical instabilities. However, the most difficult design requirements are those which are inherent in all wind energy systems. These are the requirements for all-weather operation, long service life, and low cost. In spite of these difficulties, reliable and economical wind turbine blades can be built, provided that the design is verified by structural analysis of the scope described in this paper.

For convenience, the activities of design and analysis are treated here as being separate and distinct. In reality, they are closely connected and iterative at the detail level. References 1 and 2 provide additional background information to illustrate this iterative process.



## SPECIFICATIONS

Specifications are the set of requirements which do not change as iterations of structural design and analysis take place. In general, specifications restrict the designer and establish fixed allowable conditions or "criteria" for the analyst. Thus, blade specifications must be clearly defined and should be no broader in scope than is necessary to insure that the blade is compatible with the rest of the wind turbine system. The specifications of interest to the structural analyst can be grouped conveniently into the following five categories: (1) performance; (2) site; (3) geometry; (4) loading; and (5) reliability.

Performance specifications define requirements on rotor power, annual energy output, rotor speed, and wind speeds for cut-in, rated, and cut-out operations. Site specifications would include the annual average wind speed, the cumulative distribution of wind speeds during the year, the roughness of the terrain, wind turbulence, wind shear, the elevation of the site, temperature extremes, and the seismic zone. Further information on performance and site characteristics can be found in reference 3.

Geometry specifications establish requirements for compatibility between the blade and the other components of the wind turbine system. These specifications include requirements on the size of the blade, its aerodynamic profile, definition of interfaces such as hub connection, total blade weight, allowable ranges for blade natural frequencies, and allowable deflections.

Loading specifications define sets of operating conditions which constitute the required design load cases. Often, the structural analyst will add other cases which are found to be more severe. Tables I and II indicate the scope of design load cases currently defined for large horizontal-axis wind turbines. Table I lists the 21 load conditions and additional stability conditions defined for the 2.5 MW DOE/NASA Mod-2 wind turbine system now being designed by the Boeing Engineering and Construction Company (ref. 4). Table II presents the operating conditions for 10 load cases considered during the design of the 3.0 MW GROWIAN wind turbine currently being designed by the M.A.N. firm for the West German government.

Reliability specifications are closely connected with the loading specifications since they establish lifetimes and failure modes which must be considered for each load case. They also define other possible requirements such as lightning and corrosion protection and fail-safe design.

## MATERIALS DATA

Documentation on the materials of construction is an important item which is shared by the structural designer and analyst.

This documentation generally includes the following information:

1. Selection criteria, the specific materials in the design.
2. Physical properties, such as specific gravity, thermal expansion coefficient, and corrosion resistance.
3. Mechanical properties, such as design allowable yield, ultimate, and fatigue stresses (S-N) curves, elastic moduli, ductility, and fracture toughness.
4. Quality assurance considerations such as procurement specifications, acceptance tests, process development tests, and inspection criteria.

Figure 1 illustrates documentation of fatigue allowable stresses by means of an S-N curve, in this case for a fiberglass/epoxy composite material. A curve-fit line through the test data is reduced in stress by one-third, to account for material variability and other degrading effects. The lettered points indicate calculated stresses for given load cases, which fall below the allowable lines, as required for a positive margin of safety.

Extra conservatism is required in establishing fatigue allowable stresses for blades if material deterioration by corrosion and fretting is possible. This is particularly true for wind turbine blades because of the requirement that they operate in an all-weather environment for many years while being subjected to continuous cyclic loading.

#### VIBRATION ANALYSIS

A vibration analysis is conducted to verify that the natural frequencies of the blade are within allowable ranges, to avoid amplification of periodic loads. In addition, mode shapes are defined for each natural frequency for later use in the calculation of aeroelastic loads. The scope of the vibration analysis should also include consideration of aerodynamic instabilities such as classical flutter and divergence.

Vibration mode analysis of structures like blades is usually conducted by means of finite element models and structural analysis computer codes like NASTRAN. The finite element model may be very detailed (ref. 6) if many modes are required or quite simple (ref. 7) if a few modes are sufficient. Results are presented as frequency tables (table 3, from ref. 6), Campbell diagrams (figure 2, from ref. 2), and normalized deflection shapes.

#### LOAD ANALYSIS

The objective of the load analysis is to define the forces and moments acting on cross-sections of the blade at stations

along its span. These forces and moments are technically "internal" loads which result from "external" load sources such as the wind, gravity, and inertia. Static, cyclic, and transient loads must all be calculated in order to evaluate the structural integrity of the blade. Critical static loads occur during extreme winds. Cyclic loading occurs continuously during wind turbine operation as a result of the effects of gravity and variations in wind speed across the rotor disk. Transient loads are usually critical during rapid shutdown of the machine.

Specialized computer codes are available to calculate both external and internal loads in wind turbine blades and in complete wind turbine systems (Ref. 8). Input to these cases includes blade mode shapes and natural frequencies (see VIBRATION ANALYSIS), chord and mass distributions along the blade span, airfoil lift and drag coefficients, rotor and wind speeds, blade twist and pitch angles, etc.

The load analysis is the key to determining whether or not a design meets the specified requirements. The load analysis approach should be well documented, including the load cases to be analyzed, the computer codes to be used, supporting information such as sign conventions and nomenclature, and limitations of restrictions. The load analysis is often conducted on an idealized model of the blade, and this model must be documented as well.

Figures 3 and 4 show typical results of blade load analysis. Figure 3 (Ref. 2) shows a typical spanwise distribution of maximum and minimum flatwise (out-of-plane) bending moments which occur during each rotor revolution at rated conditions. These loads are the basis of a subsequent fatigue endurance analysis for this operating condition. Figures 4 (a) and (b) (Ref. 9) show typical variations in cyclic flatwise and edgewise (in-plane) moment loads near a blade root, as the wind speed varies. Cyclic moment is defined as one-half the difference between the maximum and minimum moments during one rotor revolution. The predicted lines in Figures 4 (a) and (b) are for levels of load designated as "mean +  $1\sigma$ ", which means that approximately 84 percent of the rotor revolutions at a given wind speed are expected to exhibit cyclic loads smaller than the prediction and 16 percent would be larger. Consideration of the statistical variation in blade loads is required either in each load analysis or in the subsequent stress analysis.

#### STRESS ANALYSIS

After loads have been defined at selected blade cross-sections for each required load case, local stresses can be calculated by conventional methods. The simplest of these methods considers the blade to be a beam. This is usually sufficient for the calculation of spanwise stresses in the surface elements of the blade away from

discontinuities. Blades with internal webs and spars are treated as multi-cell airplane wings (Ref. 10) when shear stresses are important.

For increased accuracy, finite-element models of the NASTRAN type are used for the analysis of stress, particularly at critical joints. Figure 5 (Ref. 2) illustrates the complexity of a finite-element model of a blade root. Because of this complexity and the accompanying expense, finite-element modeling is usually restricted to critical segments of the blade, in order to calculate joint stresses or to verify buckling margins, for example.

Because of the requirement that wind turbine blades must function under cyclic loading in an all-weather environment for many years, special attention must be given to so-called "secondary" stresses. These are the stresses caused by discontinuities in cross-sections, transverse loads from spars and ribs, fit-up loads, etc. These stresses can contribute to fatigue failures and must be considered as primary, not secondary.

Documentation of the stress analysis procedures should include descriptions of the critical sections to be analyzed, the computer codes used, cross-sectional dimensions and properties, and appropriate stress concentration factors which account for fasteners and other discontinuities.

#### FAILURE MODE ANALYSIS

Failure mode analysis is the final step in judging the structural integrity of a wind turbine blade or of any structural component. Failure modes which must be considered include fatigue, buckling, yielding, fracture, deflection, and wear. Theoretically, the analyst can determine margins of safety by simply comparing calculated stresses with design allowable stresses (see MATERIALS DATA). In reality, engineering judgment is required because of such factors as the statistical nature of the blade loads, approximations in the stress analysis, and allowances for environmental effects, unless the latter have been included in the design allowable stresses.

Upon completion of the failure mode analysis, margins of safety are documented for all sections of the blade, with respect to each failure mode. Figure 6 illustrates this documentation for the Mod-2 blade (Ref. 4). The structural analyst then judges whether or not the requirements of the specifications have been satisfied and discusses deficiencies, if any. Finally, recommendations are made concerning any design changes or operational limits.

#### CONCLUDING REMARKS

The design and analysis of wind turbine blades is still in a state of development. Nevertheless, valid judgments as to the



structural integrity of a proposed blade will continue to depend on careful consideration of the factors described briefly herein: the specifications, materials data, vibrations, loads, stresses, and failure modes. In general, the structural integrity of a wind turbine blade is judged by the same methods which are used for many other structures. However, the difficult requirements of all-weather operation and very long life under continuous cycling demand that special consideration be given to secondary stresses, and that extra conservatism be used in setting fatigue allowable stresses.

#### REFERENCES

1. Cherritt, A. W.; and Gaidelis, J. A.: 100-kW Metal Wind Turbine Blade Basic Data, Loads, and Stress Analysis. NASA CR-134956, 1975.
2. Griffee, D. G., Jr.; Gustafson, R. E.; and More, E. R.: Design, Fabrication, and Test of a Composite Material Wind Turbine Rotor Blade. DOE/NASA/9773-78/1, NASA CR-135389, 1977.
3. Justus, C. G.: Winds and Wind System Performance. Franklin Institute Press (Philadelphia), 1978.
4. Anon.: Mod-2 Wind Turbine System Concept and Preliminary Design Report. NASA CR-(to be published).
5. Kussman, A.: Wind Rotor Load Conditions. DFVLR, Inst f. Bau. U. Konst, Stuttgart (no reference available).
6. Stahle, C., Jr.: Mod-1 WTG Dynamic Analysis. Wind Turbine Structural Dynamics, DOE Publ. CONF-771148, NASA Conf. Publ. 2034, 1978, pp. 15-29.
7. Sullivan, T. L.: Simplified Modeling for Wind Turbine Modal Analysis Using NASTRAN. Wind Turbine Structural Dynamics, DOE Publ. CONF.-771148, NASA Conf. Publ. 2034, 1978, pp. 31-37.
8. Spera, D. A.: Comparison of Computer Codes for Calculating Dynamic Loads in Wind Turbines. DOE/NASA/1028-78/16, NASA TM-73773, 1977.
9. Spera, D. A.; Janetzke, D. C.; and Richards, T. R.: Dynamic Blade Loading in the ERDA/NASA 100-kW and 200-kW Wind Turbines. ERDA/NASA/1004-77/2, NASA TM-73711, 1977.
10. Peery, D. J.: Aircraft Structures. McGraw-Hill Book Co. (New York), 1950, p. 491.

## DISCUSSION

- Q. There was much talk about all the available tools, but no discussion of the most difficult part of the analysis, the fatigue environment over the life of the machine. What are your thoughts on that?
- A. As you may recall, in the S-N curve that was shown, there were some words about spectra. The machine does not see just one cyclic load through its life which gives rise to cyclic stresses of a certain amplitude. It sees a wide variety of load cycles, and generally what is done is to categorize those cycles as to their size and the number of times they occur. This then becomes the load spectrum.

We do not yet have measured load spectrum for wind turbines, as you might expect would exist for a bomber wing or a fighter aircraft tail section. Also, we don't as yet have extensive "flight" spectra. Such data will be obtained from the Mod-OA and Mod-1 tests. However, it is necessary to account for not only the normal operating conditions, but for all of the abnormal operating conditions that make up this spectrum.

- Q. The cyclic loading is basically a statistical process. If it is measured on the Mod-0, that doesn't indicate what will happen to another machine. A better feel may be obtained, but you won't know for sure. Don't you have to approach the problem in a statistical sense?
- A. Yes. We are doing that in any load calculation. Some type of a probability of that load occurrence must be assigned. On one of the diagrams which was shown, the calculated loads were assumed to be the mean load plus one sigma, or one standard deviation. That is, 84 percent of the cycles at that particular condition of wind would be expected to fall below that load level. That will probably vary from machine to machine.

Our experience can help with certain items. For example, the number of times the machine is started and stopped can be estimated. This is a very significant load cycle. Also, the response of the Mod-OA machines to gusts tells us how often gusts of various sizes will occur.

- Q. Did the failure of the Mod-0 blade occur on the upper or lower surfaces or on both surfaces?
- A. The cracks which did occur in the root area were first seen on the low pressure side of the blade, which would be the upper side. The blade is highly twisted, so that is why it is hard to answer that precisely. However, cracks generally occurred on the low pressure side of the blade first.

TABLE 1. - LOAD AND STABILITY CONDITIONS FOR THE 2.5 MW MOD-2 WIND TURBINE SYSTEM (REF. 4).

Function		Environment								
		Normal (-40 <sup>o</sup> 120 <sup>o</sup> F)	Gust	Extreme wind	Ice	Snow	Hail	Projectile impact	Seismic	
Normal Operating	Startup	1	3					4	5	6
	Operating	2								
	Shutdown	7								
	Parked									
Operating Fault	Loss of electrical load	14	15							
	Control sys. malfunction (1.5 x rated power)									
	One tip jammed	16								
	One tip control lost	17								
	Inadvertent braking	18								
Transportation and Handling	Shipping	19								
	Handling	20								
	Erection	21								
Stability	Classical blade flutter & divergence Stall flutter Rotor/generator Tower vortex shedding				Flap/lag/torsion Rotor/tower Pitch control feedback Yaw drive control					

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TABLE II

SUMMARY OF DESIGN LOAD CONDITIONS FOR THE WEST GERMAN GROWIAN WIND TURBINE

Load Case No.	Design Condition	Wind speed		$\frac{\text{Rotor speed}}{\text{Rated speed}}$	Blade pitch	Fatigue life, cycles
		Steady	$\frac{\text{Gust}}{\text{Steady}}$			
(a) Cyclic Load Cases (fatigue stress allowables; 1.2 safety factor)						
1	Steady operation	spectrum	0	1.00	variable	$5 \times 10^8$
2	Upgust	rated	1.00	1.00	fixed	$10^4$
3	Downgust	cut-out	-0.40	1.15	fixed	$10^4$
4	Upgust	cut-out	0.60	1.15	fixed	50
5	Upgust	cut-out	0.60	0.85	fixed	50
6	Downgust	1.2 x cut-out	-0.33	1.15	fixed	50
7	Emergency stop	1.2 x cut-out	0	1.15	max rate	500
(b) Limit Load Cases (breaking stress allowables; 1.5 safety factor)						
8	Extreme wind (broadside)	60 m/s	0	0	-90 deg	--
9	Extreme wind (idling)	60 m/s	0	0.20	variable	--
10	Maintenance	46 m/s	0	0	0 deg	--

TABLE III.- CALCULATED NATURAL FREQUENCIES OF THE 100 KW  
MOD-0 WIND TURBINE SYSTEM. (REF. 6).

<u>MODE NO.</u>	<u>DESCRIPTION</u>	<u>FREQ (Hz)</u>	<u>FREQ (1/REV)</u>
1	ROTOR ROTATION	0.39	0.67
2	1ST ROTOR FLATWISE-CYCLIC	1.42	2.44
3	1ST ROTOR FLATWISE-COLLECTIVE	1.52	2.60
4	TOWER BENDING-Y AXIS	1.81	3.10
5	TOWER BENDING-Z AXIS	1.91	3.28
6	1ST ROTOR EDGEWISE-CYCLIC	2.43	4.16
7	2ND ROTOR FLATWISE-CYCLIC	3.28	5.63
8	2ND ROTOR FLATWISE-COLLECTIVE	3.64	6.25
9	SHAFT TORSION	4.00	6.86
10	TOWER TORSION	4.18	7.16
11	3RD ROTOR FLATWISE-CYCLIC	6.41	10.99
12	3RD ROTOR FLATWISE-COLLECTIVE	6.62	11.35
13	BLADE TORSION-ANTISYMMETRIC	6.76	11.60
14	BLADE TORSION-SYMMETRIC	6.78	11.63
15	2ND ROTOR EDGEWISE COLLECTIVE	7.54	12.92
16	TOWER 2ND BENDING ~ Z AXIS	8.37	14.35
17	TOWER 2ND BENDING ~ Y AXIS	8.70	14.91
18	2ND ROTOR EDGEWISE - CYCLIC	9.10	15.59

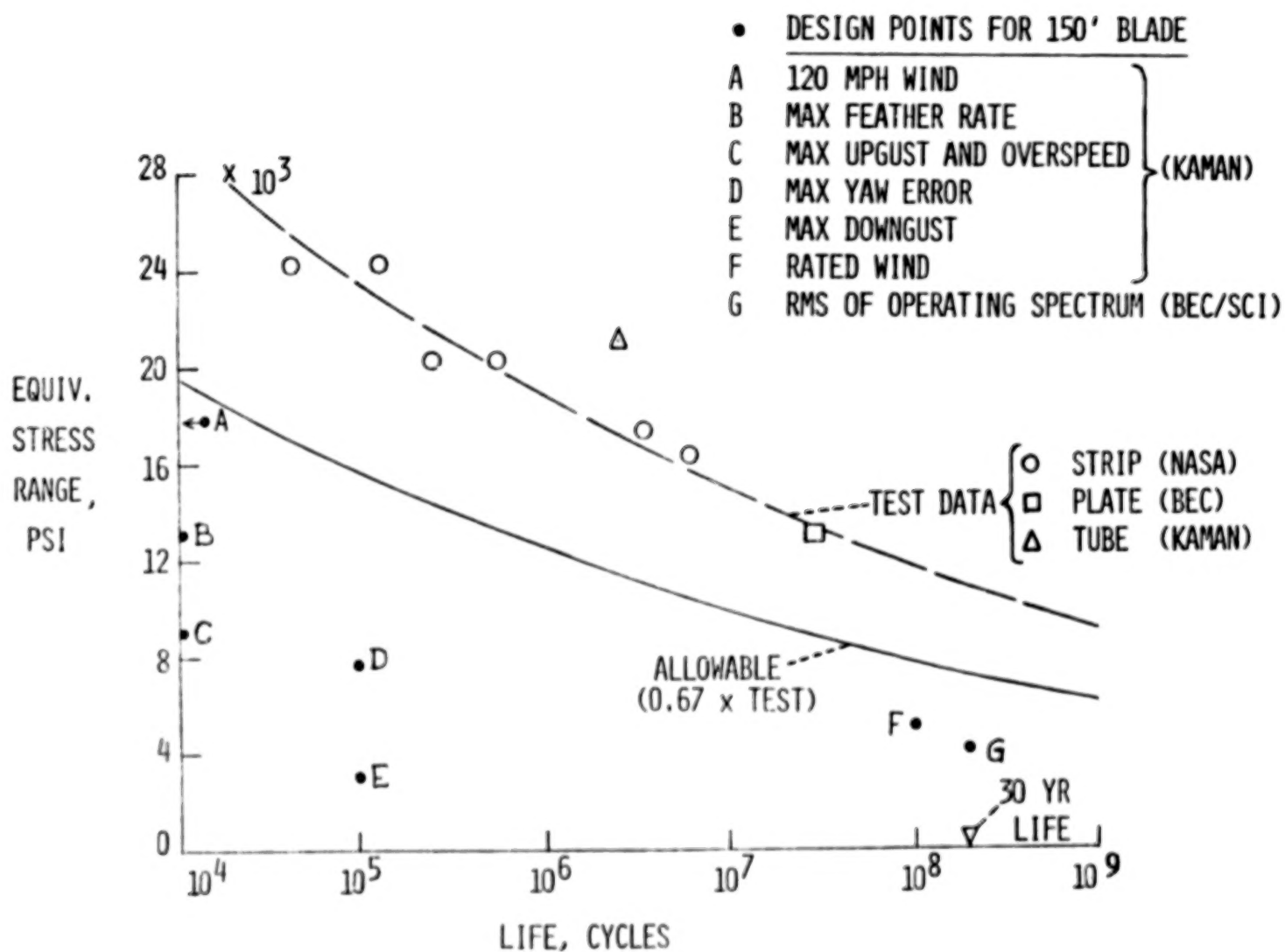


FIGURE 1. - FATIGUE DATA FOR FIBERGLASS/EPOXY COMPOSITE MATERIAL (TFT).

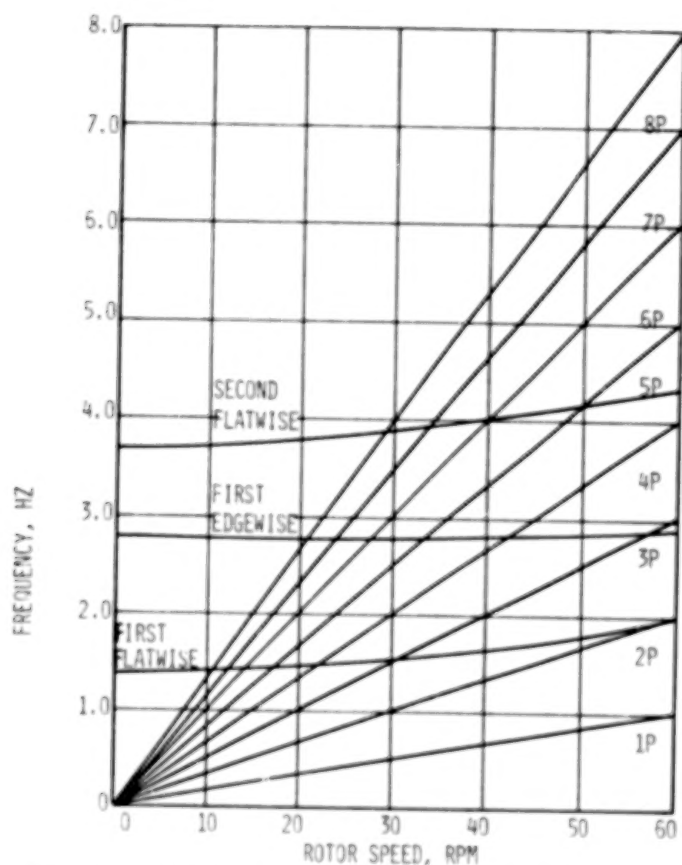


FIGURE 2. - CAMPBELL DIAGRAM OF UNCOUPLED FREQUENCIES FOR 62.5 FT. COMPOSITE WIND TURBINE BLADE. (REF. 2).

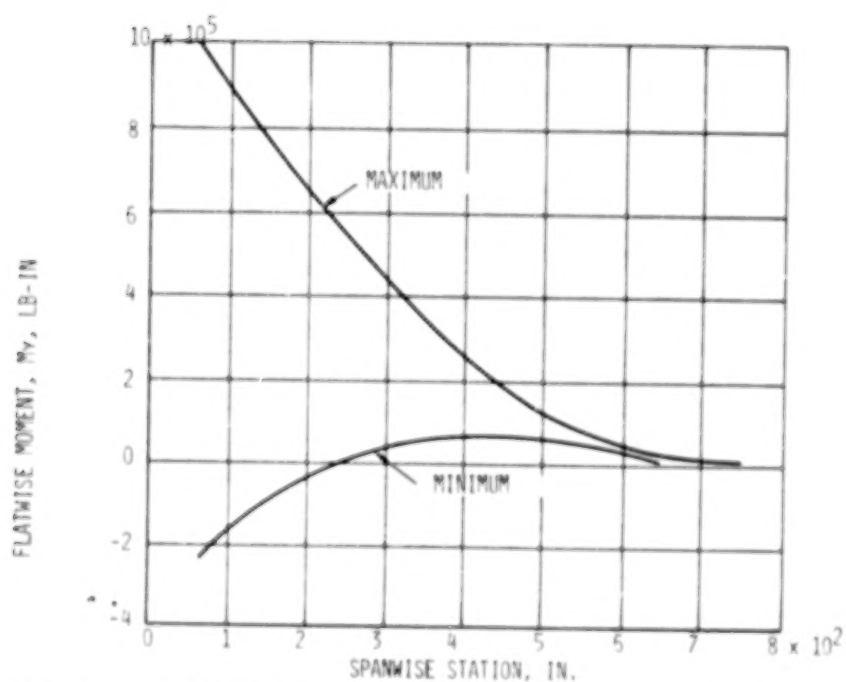
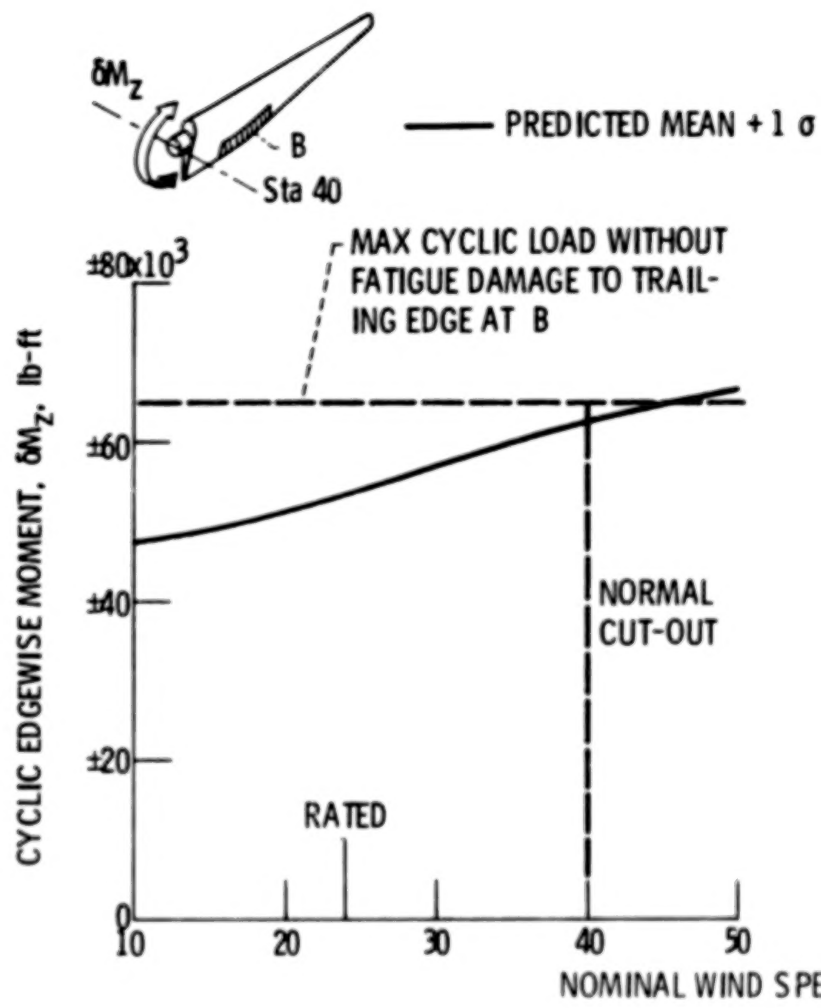
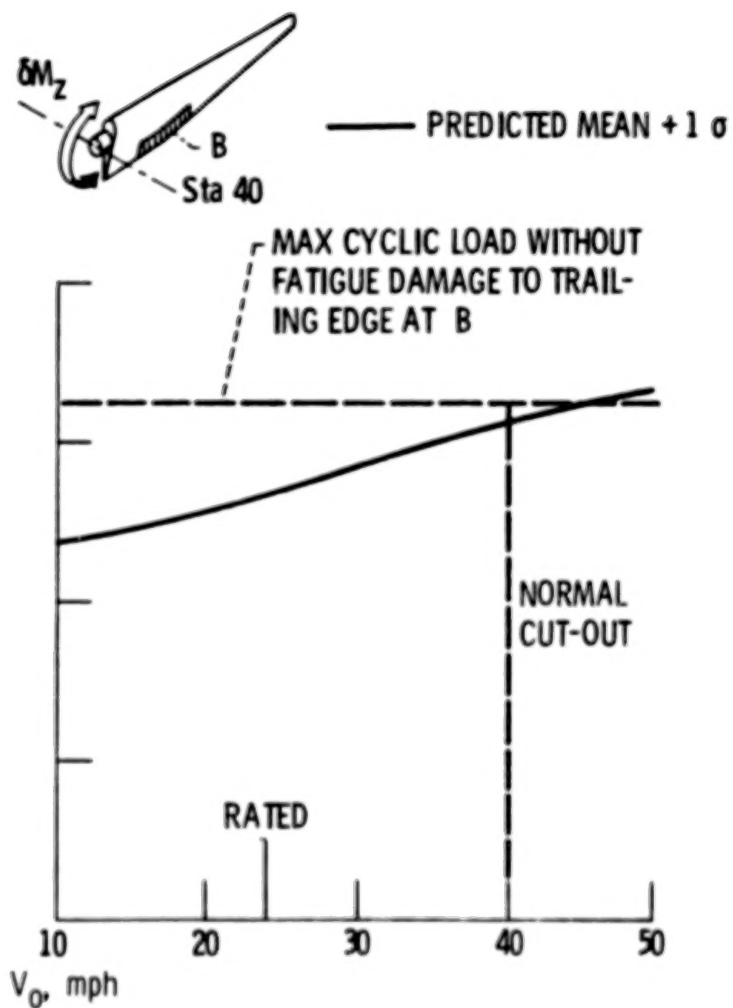


FIGURE 3. - CALCULATED FLATWISE MOMENT DISTRIBUTION IN 62.5 FT. COMPOSITE WIND TURBINE BLADE - LOAD CASE 1. (REF. 2).



(a) Flatwise bending.



(b) Edgewise bending.

FIGURE 4. - PREDICTED CYCLIC FLATWISE AND EDGEWISE BENDING LOADS FOR MOD-OA 200 KW WIND TURBINE BLADES.

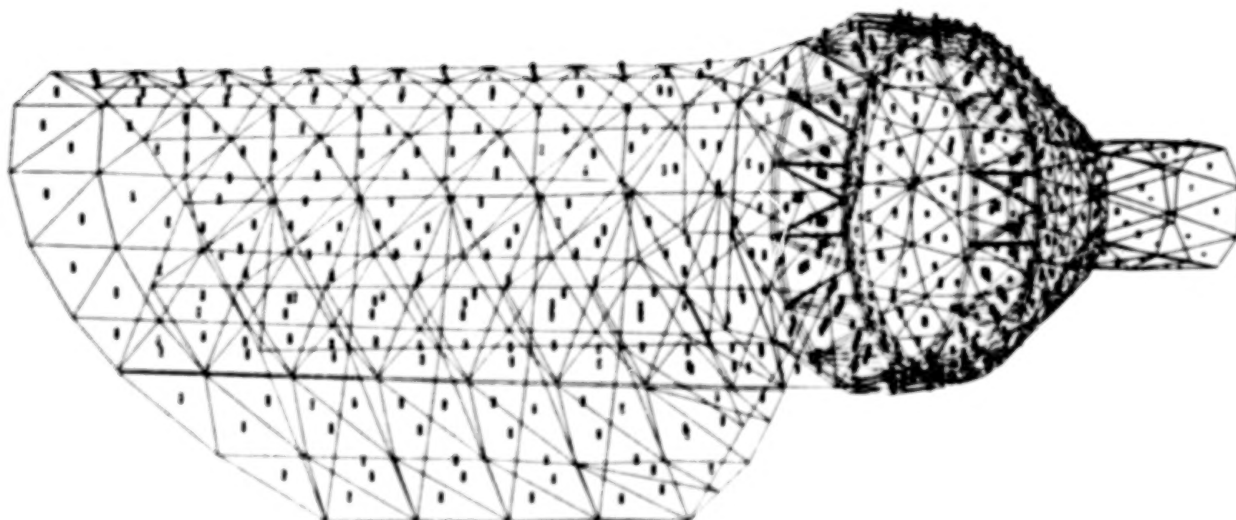


FIGURE 5. - FINITE ELEMENT MODEL OF ROOT SECTION OF  
62.5 FT. COMPOSITE WIND TURBINE BLADE. (REF. 2).

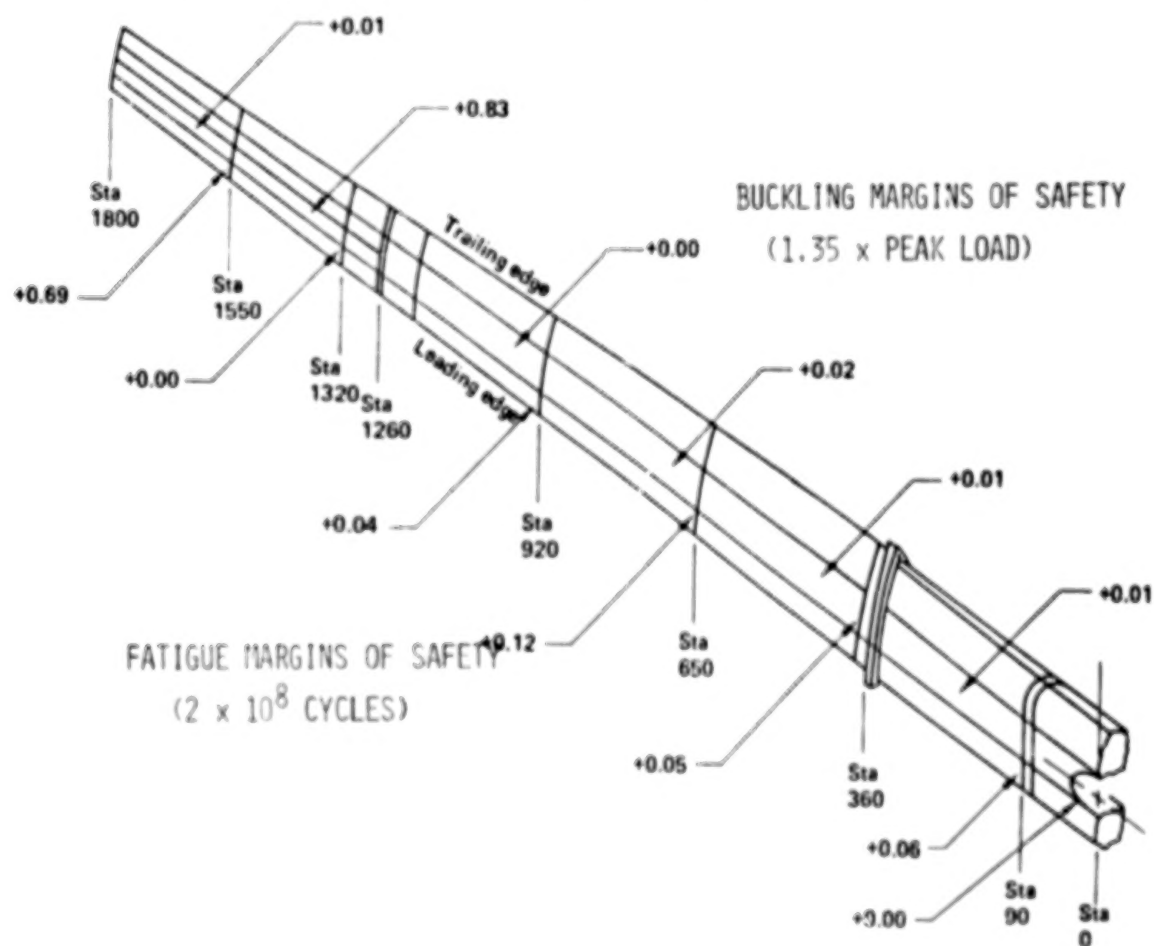


FIGURE 6. - ROTOR BLADE SKIN GAGES - MARGINS OF SAFETY FOR THE  
2.5 MW MOD-2 WIND TURBINE SYSTEM. (REF. 4).

BLADE DESIGN AND OPERATING EXPERIENCE  
ON THE MOD-OA 200 KW WIND TURBINE  
AT CLAYTON, NEW MEXICO

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SUMMARY

A 200 KW wind turbine called MOD-OA is located in Clayton, New Mexico. The MOD-OA wind turbine blade is a 60 foot long aluminum structure, similar in appearance to an airplane wing, that weighs 2,360 lbs. The blades, SN-1004 and SN-1005, accumulated over 3000 hours of operating time between November 1977 and April 1979. Signs of blade structural damage were first observed after 400 hours of wind turbine operation. The blades were removed from the wind turbine for repairs in June 1978. Repairs were completed and the blades were installed on the wind turbine for renewed operation in September 1978. Details of the blade design, loads, cost, structural damage and the blade repair are discussed.

INTRODUCTION

The U. S. Department of Energy (DOE) is responsible for wind turbine development. The management of one phase of the program-- large horizontal axis wind turbine development has been assigned to the Lewis Research Center of the National Aeronautics and Space Administration (NASA).

The overall objective of the MOD-OA wind turbine project is to obtain early operation and performance data while gaining experience by operating in a typical utility environment. The first MOD-OA wind turbine became operational at a utility site in Clayton, New Mexico in early 1978.

Because of limited funding and time allowed, a laboratory type of technology development phase of the MOD-OA project was not feasible. Instead, it was decided that wind turbine technology development would be conducted while utility operating experience was being gained.

The objectives for the development of the MOD-OA blades were (a) to test the blades on the wind turbine and not in the laboratory and (b) as technology problems were found, devise

solutions and make the necessary modification to maintain the operational status of the blades. The experience gained as a result of operating the wind turbines on a public utility is part of a planned learning process to develop better performing and lower cost blades.

The MOD-OA blades are designed and constructed in much the same way as an airplane wing. During laboratory structural acceptance tests on airplane wings, detail structural design deficiencies are often found. The deficiencies are repairable by structural modification. Usually the modifications are isolated to a particular region. These problem areas are often called "hot spots." Like airplane wings, the wind turbine blades have developed some "hot spots." As a result the blades have required some structural modifications.

Details of the MOD-OA blade design, loads and cost are discussed in this paper. During early operation of the blades, on the MOD-OA wind turbine, blade structural damage was observed. As a result, structural repair to each blade was needed. The structural blade damage and the necessary repairs are discussed in this paper.

#### BLADE SPECIFICATIONS

The blade specifications are summarized in Table I. The specifications include the blade dimensions, materials, the airfoil type and the flapwise and chordwise cantilever natural frequencies. The planform of the blade is shown in figure 1. Strain gages are located at the root end and at blade midspan. The strain gages are used to measure the flapwise and chordwise bending moments and torsion. An ice detector is located at mid span. The ice detector provides a signal, during ice build up, that initiates the shutdown of the wind turbine. A twenty-four hole bolting flange at the root end of the blade provides the mechanical interface between the blade and hub of the wind turbine. A fifty-five pin standard electrical connector provides the electrical interface. The connector provides the electrical needs for the strain gages and the ice detector.

#### DESIGN DETAILS

Figure 2 shows a cross section of the blade, taken at sta 300, also called out in figure 1. The forward portion of the cross section is called the D-spar. The aft portion of the cross section is called the trailing edge. The D-spar is a heavier and stronger portion of the blade and as a result it carries the majority of the applied loads. Angle stringers and ribs, shown in figure 2, are needed to prevent panel buckling, of the .08 inch and the .31 inch thick outer skins, due to compressive loads.



Detail A, called out in figure 2, is shown in figure 3. A typical method of attaching the angle stringers to the D-spar skin and rib is shown in figure 3. The steel Hilok fastener is an aircraft type of high strength bolt. Close tolerance holes must be prepared so that there is an interference fit between each fastener and the hole. The interference fit allows the Hilok fastener to carry high shear loads. Aluminum rivets are typically used in the trailing edge portion of the blade.

Figure 4 shows the design details of the root end of the blade. A steel cylindrical tube slides through the rib at station 48 and is bolted to the rib at station 81.5 and is bolted to the web near station 81.5. The flange at station 31.75 provides the mechanical interface to the hub of the wind turbine.

## LOADS AND ANALYSIS

Table II summarizes the maximum safe, or "red line," operating blade bending moments allowed during operation. Blade loads data, taken during operation, show that the loads are generally at or below the "red line" values. However, overloads have infrequently been observed during a yaw maneuver and during an emergency shutdown of the machine. During an emergency shutdown, the blades are pitched at 4°/sec. causing a rapid decrease of the rotor speed.

The structural analysis for the MOD-OA wind turbine blade is identical to the standard methods used for aircraft wings (refs. 1 and 2). Further discussion of operational blade loads and analysis is found in ref. 3. Several computer codes are being used to calculate blade loads during simulated operating conditions (ref. 4).

## MOD-OA BLADE DESIGN DRIVERS

The DOE/NASA research wind turbine is called MOD-O, and it is located in Sandusky, Ohio (ref. 3). Soon after the MOD-O wind turbine was operating the DOE requested NASA to begin fabricating the first MOD-OA wind turbine. It was necessary that the first MOD-OA machine be completed within certain cost and time requirements. In order to meet the cost and schedule requirements, it was necessary to use the basic MOD-O aluminum blade design for the MOD-OA blades.

During early operational experience with the MOD-O wind turbine, the measured MOD-O blade loads were higher than the loads used to design the blades (refs. 1 and 3). As a result, it was decided to redesign the MOD-O blades, to carry higher loads while maintaining the 50,000 hour life requirement.

During early operational experience with the MOD-OA wind turbine at Clayton, New Mexico some blade design deficiencies were found. Also, certain wind turbine operating conditions were found that imposed blades loads in excess of the design allowables. Structural design changes were devised, and structural modifications were performed on the blades, to correct the design deficiencies. The design changes and structural modifications will be discussed in the section entitled "The Clayton Experience."

### COST & COST DRIVERS

Table III summarizes the costs for MOD-OA blades. It is noted that the reduction in cost for the blades (S/N 008, S/N 009) is due to more efficient assembly procedures. Also, the costs shown in Table III are actual costs with no adjustment for inflation.

The process used to fabricate each blade is labor intensive. Each blade is made up of many individual parts, each requiring a number of hand operations during most phases of fabrication. Examples of this highly labor intensive fabrication and assembly process are as follows:

1. Brake forming of the individual D-spar .25 and .31 thick skin panels
2. Hand trimming and fitup of each individual D-spar panel and trailing edge panel
3. Individually drilling, reaming and deburring the majority of the 14,000 holes for fasteners

Additional details on fabrication of the blades are contained in reference 5.

Because a limited number of the MOD-OA blades were to be built, inexpensive wood assembly fixtures were used for the assembly. The tooling required to reduce the labor time was not economically practical for the few blades that were fabricated.

### THE CLAYTON EXPERIENCE

On November 30, 1977 the MOD-OA wind turbine was operated in Clayton, N.M. for the first time. Operational checkout of the wind turbine, at Clayton, was conducted by NASA from December 1977 through February 1978. Information on the equipment used to perform operational checkout of the wind turbine is found in reference 5. The wind turbine was turned over to the Town of Clayton Light and Water Plant on March 6, 1978, for routine operation during a two year experimental period. The operational experience gained during the first ten months of utility operation in 1978 is described in reference 6.

In late March 1978, Utility Company personnel reported to NASA that a creaking noise was emanating from the blades. They also reported the appearance of a gray discoloration around several protruding head fasteners. The fasteners were located between station 48 and 80 along the joint connecting the trailing edge to the D-spar. By late March 1978 each blade had accumulated about 400 hours of operation, which is equivalent to about one million load cycles.

In April 1978, NASA inspected the blades at Clayton. As a result of the inspection, two broken fasteners and several loose fasteners were found on one of the blades. These fasteners were located between station 48 and 80 along the joint connecting the trailing edge to the D-spar. Because structural damage to the blade was found, the blades were removed from the wind turbine in June 1978. The blades were then sent to the Lewis Research Center in Cleveland, Ohio for a more thorough inspection of the structure. At the time the blades were removed from the wind turbine in June 1978, the blades had accumulated 1,124 hours of operation at 40 rpm or  $2.7 \times 10^6$  load cycles. It is important to note the accumulation of a significant number of load cycles ( $2.7 \times 10^6$ ) over just a few months of wind turbine operation.

As a result of the blade inspection at the Lewis Research Center, two blade design deficiencies were found.

One design deficiency was located along the line of fasteners joining the trailing edge skin to the D-spar, shown in figure 5a. It was determined that the joint, shown in figure 5a, could not carry the applied shear loads. The shear loads occur primarily due to the weight of the blade. As a result broken and loose fasteners were found, and cracks were found in several angle stringers located in the root end end of the D-spar. This design deficiency was corrected by adding doublers, on the exterior of the blade, between station 48 and 100, as shown in figure 6.

The second design deficiency was the bearing interface, located between the aluminum rib, at station 48, and the steel root end fitting, shown in figure 5b. It was observed, during the blade inspection at the Lewis Research Center, that the steel root end fitting had rubbed on the aluminum rib during operation. This rubbing action caused excessive wear of the aluminum rib. The wear resulted in a radial clearance of 0.14 inches between the root end fitting and the rib as shown in figure 5b. Because of the large radial clearance at sta 48, a bending moment, larger than the design allowable, was applied to the rib at station 81.5. The high bending moments applied to the station 81.5 rib caused the rib to crack and also caused the 0.25 thick D-spar skin to crack.

As a result, doublers were added to the blade exterior at station 81.5, as shown in figure 6. A special bearing was designed, as shown in figure 7, to reduce the wear of the aluminum rib at station 48.

The two structural design deficiencies described above are often referred to as structural "hot spots" in the aircraft industry. As in the case of newly developed aircraft, it is not unusual to find hot spots during the early operation of a new wind turbine blade structure. The structural damage can be classified as a hot spot because the damage occurred over a short spanwise portion of the blade, between station 48 and station 100.

The structural modifications to the blades were completed at the Lewis Research Center, and the blades were returned and installed on the Clayton wind turbine in September 1978.

In April 1979, the blades had accumulated about 3000 hours of operating time or  $7.3 \times 10^6$  load cycles. The blade structure, outboard of station 100, appears to be in good condition and shows no evidence of structural degradation.

#### ALTERNATE MODIFICATIONS

Figure 8 shows an alternate method for securing the steel root end fitting to the blade. This modification is currently being made to the MOD-O aluminum wind turbine blades. This design eliminates the need for a bearing at station 48. A new steel rib, installed at station 69, carries load from the new root end fitting directly into the D-spar. The new rib at station 69 eliminates the need for the rib at station 81.5. As a result, no structural modification to the heavily damaged rib at station 81.5 was needed. When the blades are complete, they will be installed and operated on a MOD-OA wind turbine. Operational experience with the newly modified blades will allow NASA to assess the structural integrity of the new design.

#### CONCLUSIONS

Periodic inspection of the MOD-OA blades has resulted in the detection of structural damage in the early stages.

The blade structural damage found in June 1978, was repaired and the blades were put back into operation on the MOD-OA wind turbine during September 1978. These repairs have substantially improved the service life of the blades to over 3,000 hours, as of April 1979.

The nominal cost for the six MOD-OA wind turbine blades, purchased by NASA, is \$100/lb. This cost is primarily due to labor intensive fabrication procedures and numerous parts.

#### REFERENCES

1. Cherritt, A. W. and Gaidelis, J. A.; "100 kW Metal Wind Turbine Blade Basic Data, Loads, and Stress Analysis, DOE/NASA/9235-75/1
2. Peery, David J., "Aircraft Structures" McGraw-Hill Book Co., 1950
3. Linscott, B. S.; Glasgow, J.; Anderson, W. D. and Donham, R. E.; "Experimental Data and Theoretical Analysis of an Operating 100 kW Wind Turbine, DOE/NASA/1028-78/15; NASA TM-72883, January 1978.
4. Spera, D. A.; "Comparison of Computer Codes for Calculating Dynamic Loads in Wind Turbines", DOE/NASA/1028-78/16 NASA TM-73773, Sept. 1977
5. Hunnicutt, C. L.; Linscott, B.; and Wolf, R. A.; "An Operating 200 kW Horizontal Axis Wind Turbine", DOE/NASA/1004-78/14; NASA TM-79034, May 1978
6. Glasgow, J. C.; and Robbins, W. H.; "Utility Operational Experience on the NASA/DOE Mod-OA 200 kW Wind Turbine", DOE/NASA/1004-79/1; NASA TM-79084, February 1979

#### DISCUSSION

- Q. How much did the blade weight increase due to structural modifications?
- A. About 100 lbs. of structure was added to each blade.
- Q. Did each blade experience similar damage and were the structural modifications to each blade similar?
- A. Both blades experienced similar damage. However, the damage to blade 100 was more severe. As a result the structural modifications to blade 100 were more extensive in detail. However, from outward appearance, the modifications to each blade are nearly identical.
- Q. Does the Beryllium-Copper ring installed in the aluminum rib at station 48 cause a correction problem due to the two dissimilar metals?
- A. There is a potential galvanic corrosion problem if the two materials make intimate contact in a moist environment. However, the Beryllium-Copper ring is adhesively bonded to the aluminum rib. The adhesive provides an insulated barrier that separates the two parts, thus preventing galvanic corrosion.
- Q. Is it possible that moisture, condensed inside the blade, will freeze during cold weather and cause blade imbalance?
- A. It is likely that small quantities of water will freeze inside the blade. Drain holes, located at several places on each blade, prevent significant accumulations of water. During operation at Clayton, N.N., we have not recorded any blade imbalance attributable to ice inside the blade.



TABLE I. - MOD-0A BLADE SPECIFICATIONS

DIMENSIONS

LENGTH	59.9 FT
TIP CHORD	1.5 FT
ROOT CHORD	4.5 FT
CHORD TAPER	LINEAR
TWIST	34°

MATERIALS

ALUMINUM	2024
STEEL	4340

AERODYNAMIC

AIRFOIL	NACA	23000
SOLIDITY		38

STRUCTURAL DYNAMICS

FIRST FLAP	1.5 Hz
FIRST CHORD	2.9 Hz

INSTRUMENTATION

STRAIN GAGES:

LOCATION

ROOT END (STATION 40)  
MIDSPAN (STATION 370)

MEASUREMENT

FLAP & CHORD BENDING  
TORSION

ICE DETECTOR

SHUTDOWN WTG IF ICE IS GREATER THAN  
0.020" THICK

MECHANICAL INTERFACE

CIRCULAR BOLTING FLANGE  
24 - 5/8" DIA. HIGH STRENGTH BOLTS

ELECTRICAL INTERFACE

MS CONNECTOR, 55 PIN

BLADE WEIGHT - 2350 LB

TABLE II. - MOD-0A BLADE LOADS  
NASA "RED LINE" BENDING MOMENTS

	MAXIMUM BENDING MOMENT (FT-LBS)	CYCLIC BENDING MOMENT PEAK TO PEAK (FT-LBS)
FLAPWISE, STA. 40	200,000	130,000
CHORDWISE, STA. 40	85,000	130,000

TABLE III. - MOD-0A BLADE COSTS

SERIAL NUMBER	DELIVERY DATE & LOCATION	\$/BLADE	\$/LB.
1004 & 1005 1006 & 1007	11-77 CLAYTON, N. M. 5-78 CULEBRA, P. R.	252,000	107.
1008 & 1009	3-79 BLOCK ISLAND, R. I.	22,000*	93.*

\* \$28,000/BLADE FOR STRUCTURAL MODIFICATIONS NOT INCLUDED.



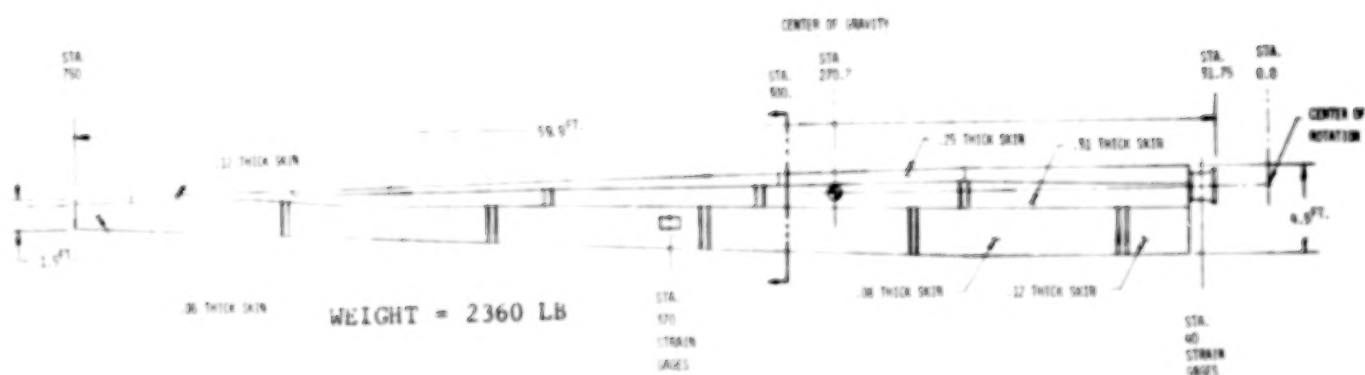


Figure 1. - MOD-OA blade configuration - planform.

CS-79-1931

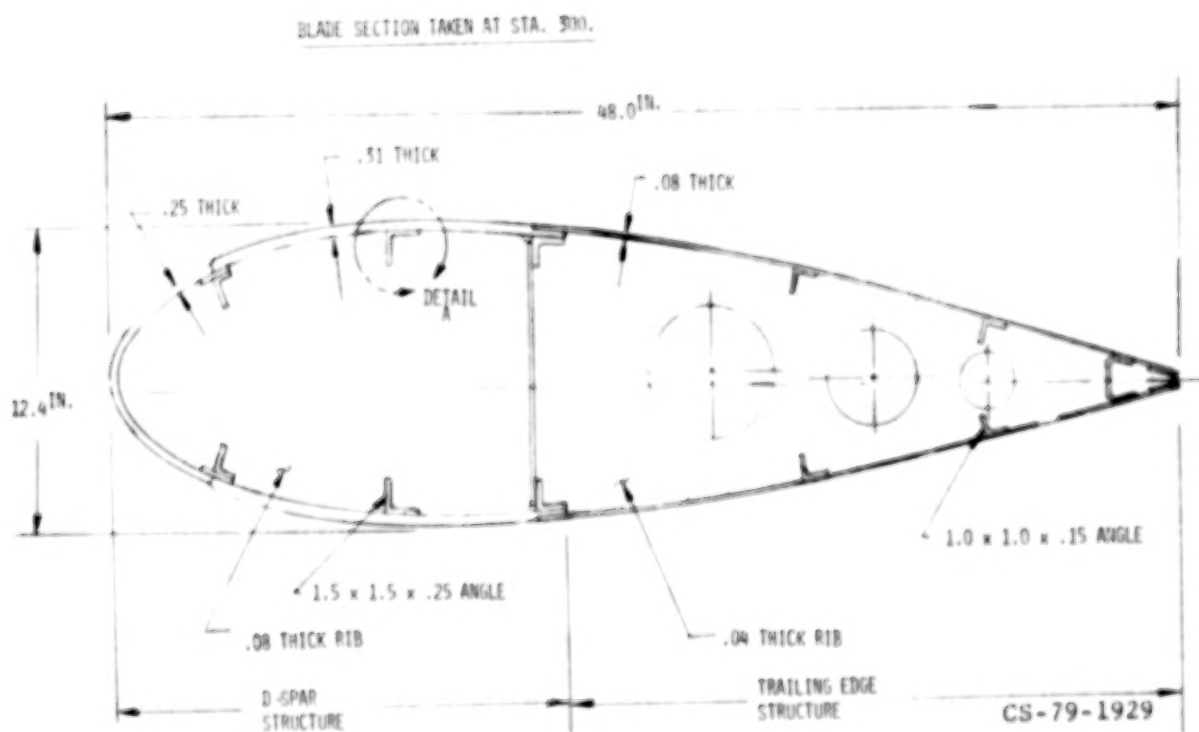


Figure 2. - MOD-OA blade typical cross section.

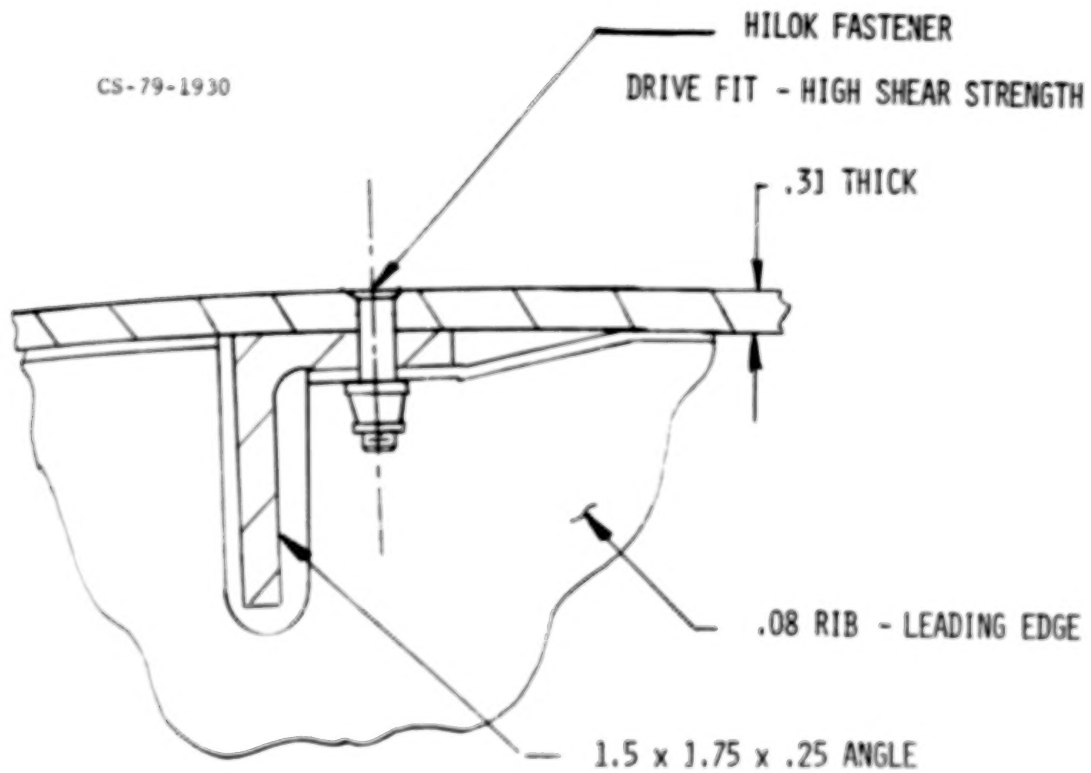


Figure 3. - MOD-OA blade fastener detail A.

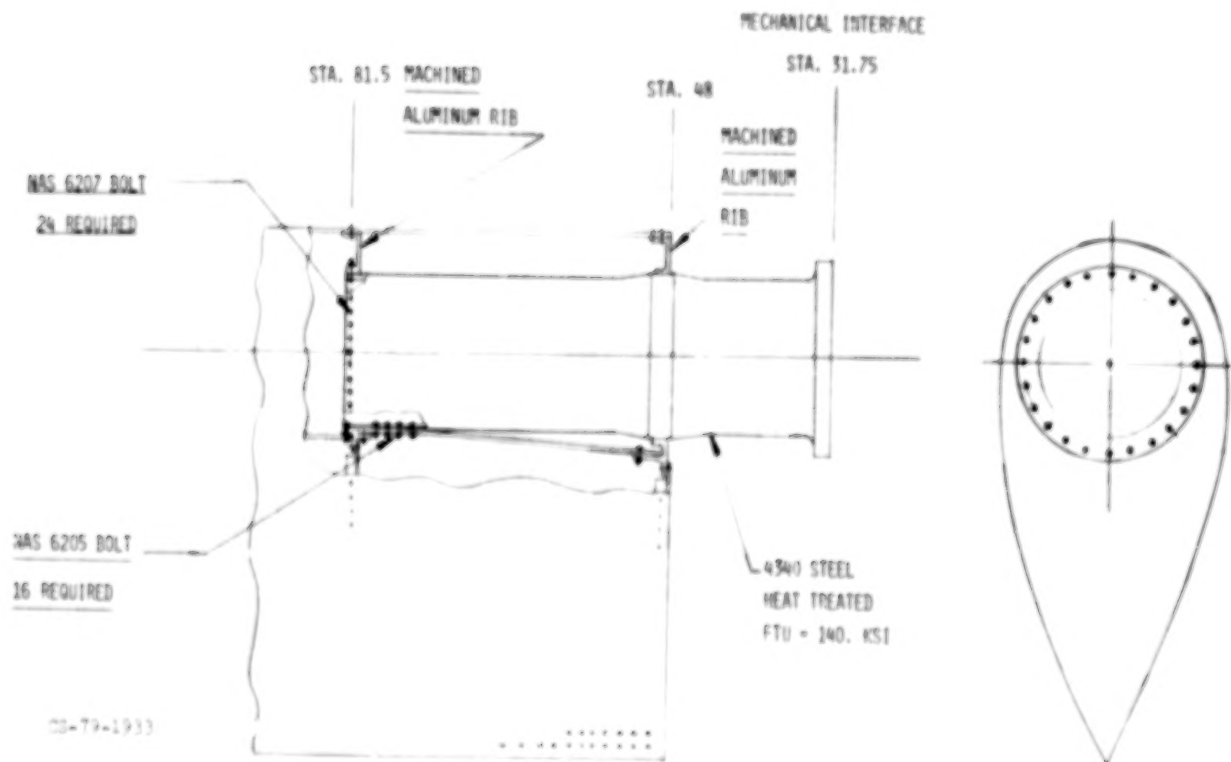


Figure 4. - MOD-OA blade root end details.

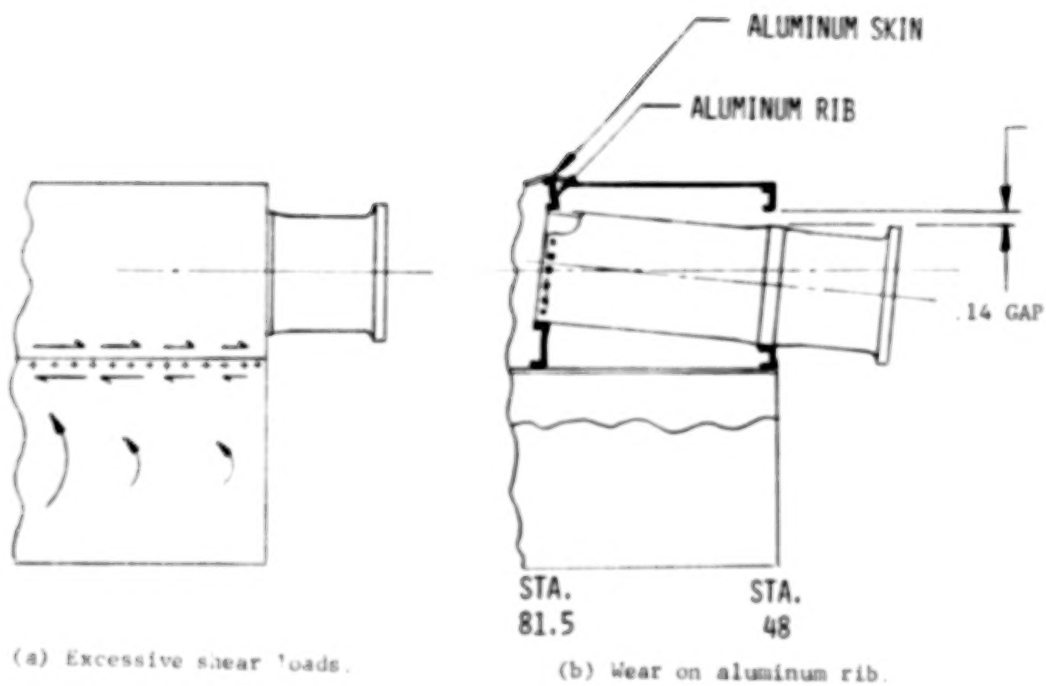


Figure 5. - MOD-0A detail design deficiencies.



Figure 6. - Final installation of doublers on the low pressure side of blade SN-1005.

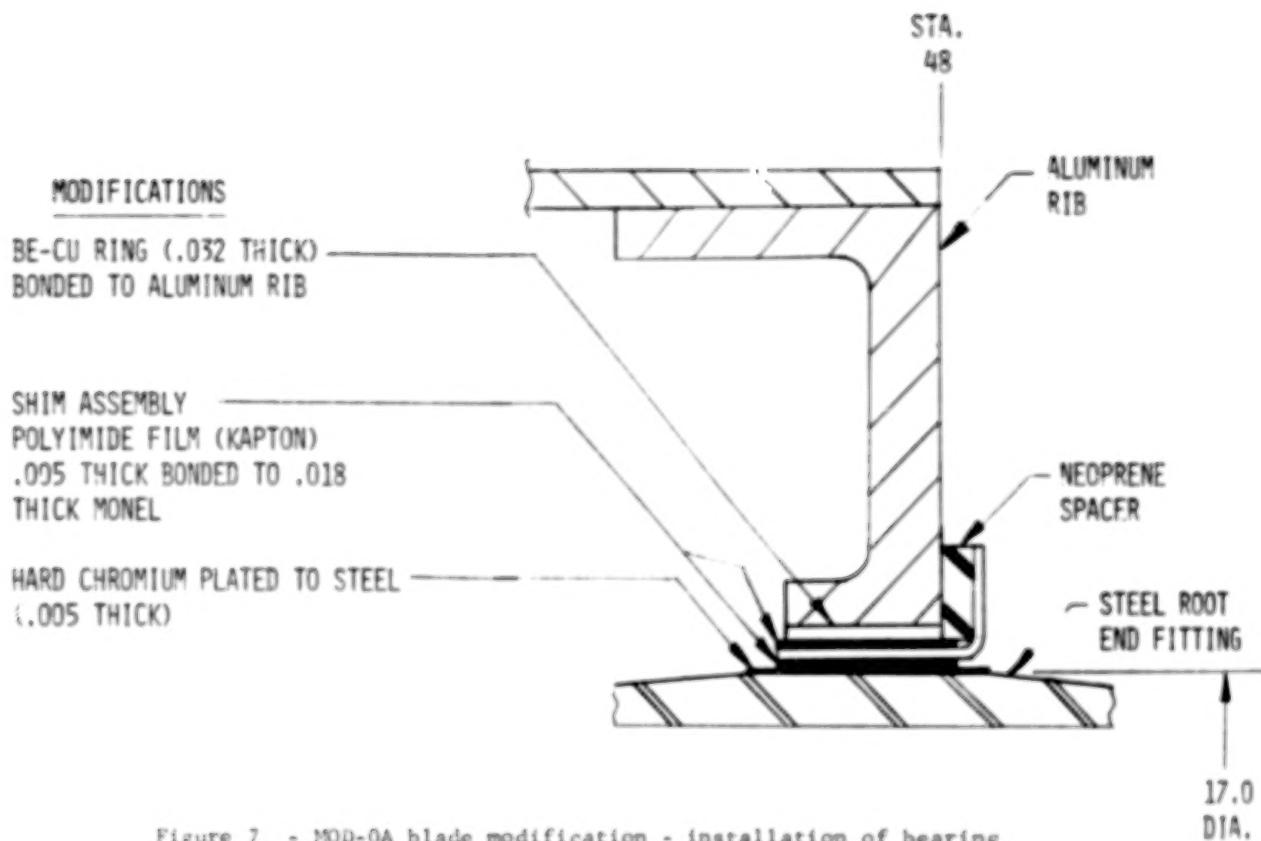


Figure 7. - MOD-OA blade modification - installation of bearing.

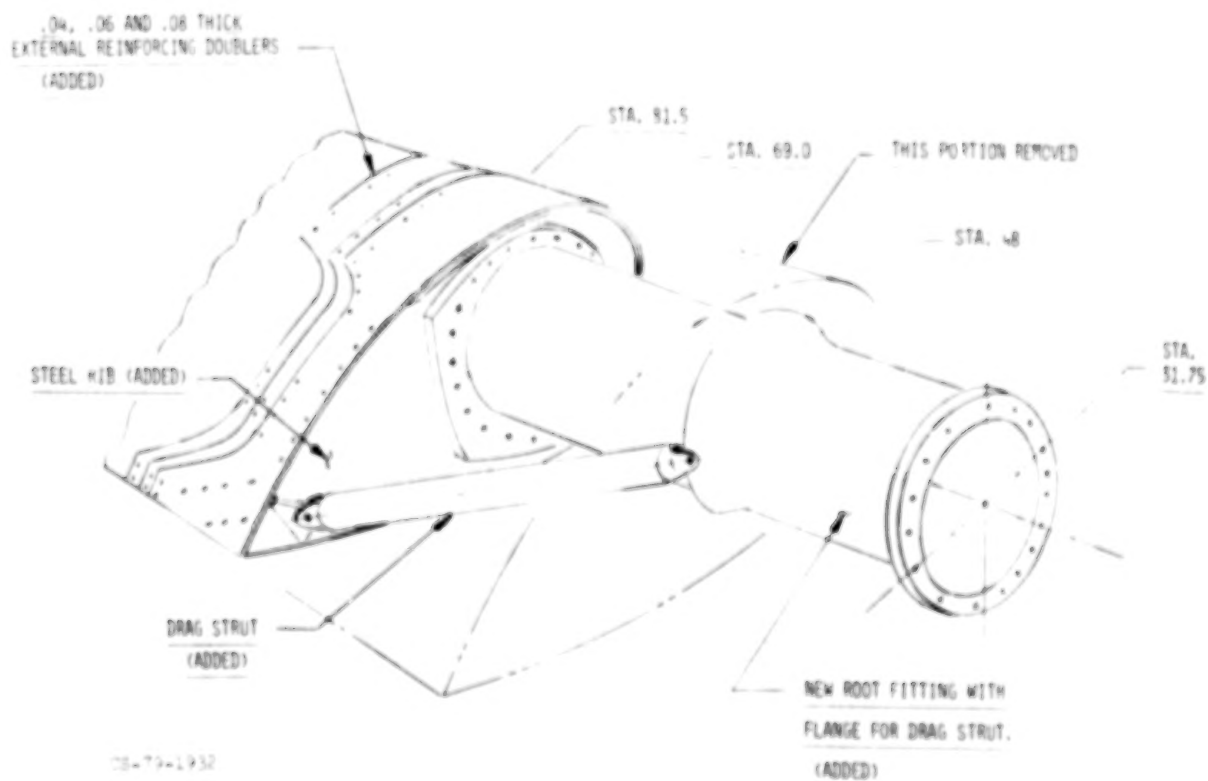


Figure 8. - MOD-OA blade. Alternate modification.

EVALUATION OF AN OPERATING  
MOD-OA 200 kW WIND TURBINE BLADE

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The Mod-OA wind turbine blades, manufactured by Lockheed Aircraft Service Company (LAS), Ontario, California, are now operating in Clayton, New Mexico. These blades, rotated for the first time on November 30, 1977, establish the Mod-OA as the first wind-driven generator in 35 years to be continually tied into an electrical power system that services a community. Two additional sets of Mod-OA blades have become operational on the Island of Culebra, Puerto Rico, and Block Island, Rhode Island.

Blade design follows that of the Mod-O wind turbine built for NASA. The Mod-OA wind turbine blades are geometrically the same as the Mod-O blades. Structural modifications recommended by Lockheed to extend the fatigue life of the Mod-O blades and NASA's experience with the Mod-O unit influenced the design of the Mod-OA turbine blade structure; so did cost and schedule constraints.

Operating limits were determined from analyses and Mod-O experience. No tests were made to corroborate the many assumptions necessary for fatigue analyses (the behavior of structural details). It is generally the practice in fatigue analysis of aircraft structures (both fixed and rotary wing) to corroborate analysis assumptions with tests. Fatigue damage of the Mod-OA blade structure during normal operation might accumulate at an unexpected rate. Appropriate caution in the form of frequent inspections and corresponding repairs (if necessary) was therefore recommended.

## DESCRIPTION OF THE 200-kW WIND TURBINE SYSTEM

The 200-kW wind turbine is a two-bladed, horizontal-axis, rotor system driving a synchronous electric generator through a step-up gear box located within a nacelle. The nacelle is mounted on top of a 100-foot tower as shown in Figure 1 with the rotor located downwind from the tower. The 200-kilowatts rated power output of the wind turbine is achieved at a turbine rotor speed of 40 rpm and a rated wind speed of 18.3 mph. The rated wind speed is defined as the lowest wind speed at which full power is achieved. The wind turbine power output, as a function of wind speed, is regulated by varying the pitch angle of the blades. At wind speed below cut-in and above cut-out the rotor blades are placed in a feathered position and no power is produced. The cut-in wind speed, defined as the lowest wind speed at which power can be generated, is 6.9 mph. The cut-out wind speed, defined as the lowest wind speed at which wind turbine operation would result in excessive blade stress, is 34.2 mph. All of these wind speeds are measured at a 30-foot elevation.

In the gear box, the shaft rpm is increased from 40 to 1800 rpm. A high-speed shaft connects the gear box to the 200 kW alternator. The drive train assembly is enclosed in a fiberglass nacelle for environmental protection. The nacelle and rotor assembly are positioned at the top of a tower to provide the necessary blade tip to ground clearance. A hoist provides access to the tower. The onsite controls and electrical switchgear are housed in the control building at the base of the tower.

The yaw drive permits rotation of nacelle/blades to maintain proper alignment with the wind. Rotation is achieved by driving a large bull gear with two pinion gears. The two pinion gears, which are preloaded against each other to increase torsional stiffness, are driven by separate motors and yaw drives. If necessary, yaw control can be achieved by using only one unit. The yaw rate, 1/6 rpm, is operational whenever the wind speed exceeds the cut-in wind speed of the wind turbine.

The torsional stiffness of the tower-nacelle interface is further increased by activating three yaw disk brakes. Even during the yawing motion, some

brake pressure is applied to damp out any torsional oscillations by maintaining a drag force. Once the machine has aligned itself to the wind, this brake pressure is increased to the maximum.

The function of a fluid coupling on the high-speed shaft is to damp out the power oscillations resulting from the continuously varying wind velocity that the blades must withstand due to the tower shadow and the wind shear effects.

#### BLADE DESCRIPTION

In many aspects, the blades are similar to an airplane wing: they contain leading and trailing edge structure, formers, stringers, ribs, webs, and skin. However, the length of each blade (62.5 feet), the taper, twist, and contour parameters it must maintain coupled with the balance, weight, and flex requirements for symmetrical blades, make them unique. All components were tested for chemical and physical properties to ensure against impurities. In addition to the required test certifications, a copy of the actual test results accompanied each certification.

Before assembly was started, the blade fixture was boresighted and adjusted to ensure contour, taper, and rigidity at all stations. The same check was performed at least three times a week during actual blade assembly.

The brake-formed leading edge, assembled in sections, serves as the base for installation of the D-spars, formers, stringers and ribs. Once installed in the blade assembly fixture, the leading edge is drawn tight against aluminum sheets fastened to the jig frame on one side and ribs on the other. Stability is ensured by use of turnbuckles and a strap that is secured to the concrete floor.

Before any adjustment of the leading edge skin is attempted, each skin segment is aligned to chord lines marked on the jig and then boresighted adjustments are made as required and the first skin segment of the leading edge is trimmed and spliced together with the second skin segment, etc., until the leading edge is one complete assembly. D-spars and ribs are added and secured to the leading edge by Hi Loc fasteners. Formers over D-spars, and stringers on both

sides of the leading edge and the formers, give additional support to the blade. Thick aluminum skins, varying from 3/16-inch just aft of the blade root to 3/64-inch at bladetip, are attached to the ribs which run the length of the blade. Except for the steel blade root fitting, all components are constructed of heat-treated 2024T3 aluminum.

All structural components are wet-sealed at assembly, and frequent inspections are made to ensure an airtight condition exists. Five hollow tubes, one in the apex of the leading edge at the blade root, one centered on ribs at the root segment; and three attached at the blade tip, permit weights to be added or removed to balance each set of blades. In addition, throughout the entire length of the blade (approximately every 22 inches) weighted tubes and solid bars are installed in the leading edge to maintain section and segment structural balance.

Strain gages, installed in the blade root and in the blade midsection, enable monitoring of flap bending, in-plane bending, and torsion moments during operation. The gages are epoxy-sealed and all wires secured to a terminal board and then, by clamps, to the ribs and blade root.

Each blade (see photo) was tested for (a) deflection and vibration, (b) weight and balance, (c) strain gage accuracy, and (d) X-rayed for defects. Each set of blades was given deflection and vibration, weight and balance, and symmetry checks.

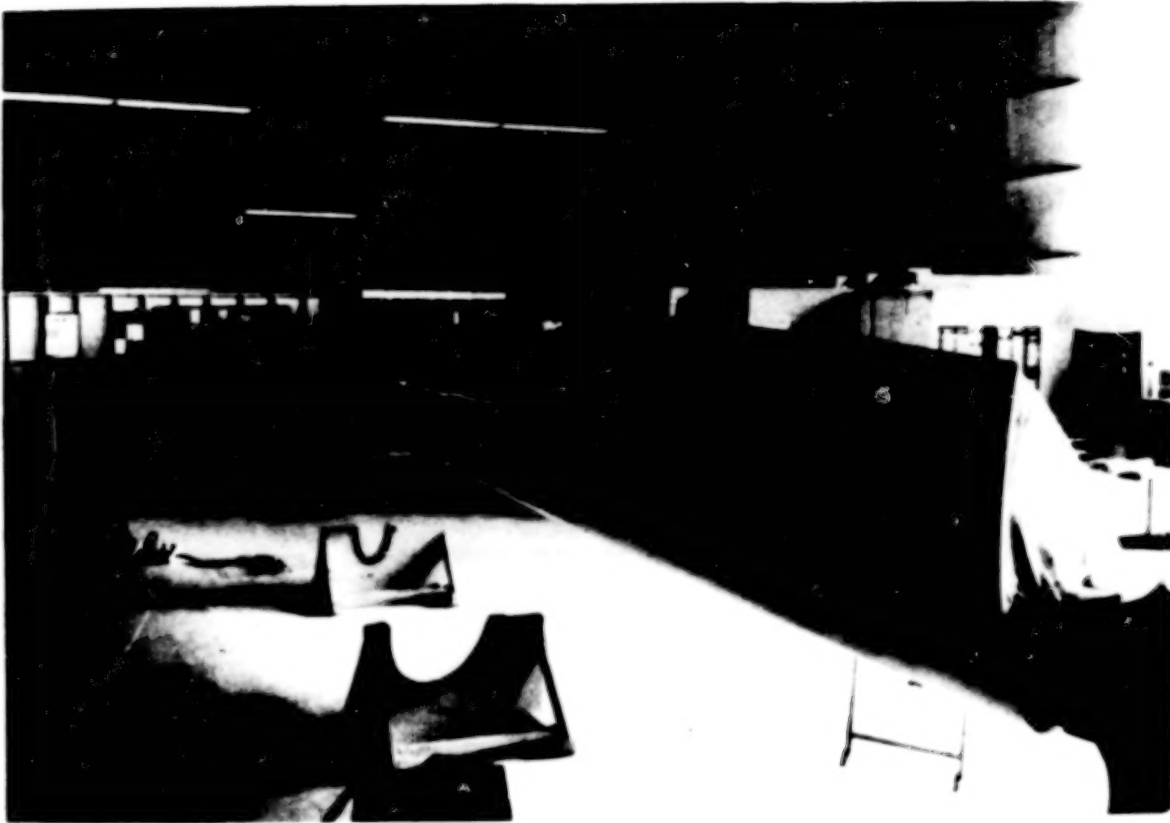
#### MOD-0 TEST/ANALYSIS EXPERIENCE

NASA LeRC selected a 100-kW WTC as being large enough to assess technology and solve engineering problems of large (1 - 3 megawatts) WTC's and yet maintain costs within the available project budget. The test program provides engineering data needed to determine whether the technology for wind energy can be used to create machines that will help meet the nation's energy needs at costs that are competitive with other systems.

Experimental test data have been correlated with analyses of turbine loads and complete system behavior of the ERDA-NASA 100 kW Mod-0 wind turbine generator over a broad range of steady state conditions, as well as during transient conditions.



MOD-0A Turbine Blade - LAS Ontario, Calif.



The Lockheed California Company, designer and fabricator of the Mod-0 metal blades, was funded under Contract NAS 3-20036 by the NASA-Lewis Research Center to evaluate the test data and conduct structural analyses of the wind turbine rotor blade to provide:

- Task I - Fatigue Analysis
- Task II - Analysis of Wind Velocity Measurement Test Data
- Task III - Correlation of Analytical with Actual Loads Data
- Task IV - Potential Structural Blade Modification

Lockheed applied two different analytic computer programs to determine loads for correlation with measured data supplied by NASA: Lockheed's WINTUR (WIND TURbine) program, a quasi-steady fully-coupled analysis method (a brief description of the method is included in reference 3); and an adaptation of Lockheed's REXOR-WT (Revised and Extended rotOR-Wind Turbine) program.

Loads computed by the WINTUR program were used to calculate stresses used in the fatigue analysis, Task I. The test conditions for which correlation was shown were:

- 40 rpm and 100 kW
- 40 rpm and zero power
- 30 rpm and zero power
- 20 rpm and zero power
- Emergency feathering

#### MOD-0 CONFIGURATION EFFECTS ON ROTOR BLADE LOADS

Three sequential configuration evolutions have resulted in the current Mod-0 wind turbine. For each of the three configurations, the wind turbine was operationally tested in a similar wind environment. The purpose of the tests was primarily to compare rotor blade loads as a function of structural configuration, while attempting to maintain an identical wind environment.

- Configuration I - The wind turbine tower was configured with stairs and rails. A single yaw drive is installed between the tower and the nacelle as described in reference 1.
- Configuration II - The wind turbine tower stairs and rails are removed. The single yaw drive between the tower and nacelle is retained.
- Configuration III - The wind turbine tower stairs and rails are removed. A mechanical lock (yaw keeper) is installed between the nacelle and tower structure. The yaw keeper provides much higher torsional stiffness in yaw rotation than the single yaw drive. This was mechanically incorporated by design and installation of a dual yaw actuation drive combined with a brake system.

Synoptically, the results that these configuration changes achieved on the wind turbine blade root bending moments (measured at 40 inches from the shaft center line) are summarized on table 1.

TABLE 1. BLADE BENDING MOMENTS MEASURED DURING OPERATION IN CONFIGURATIONS I, II AND III COMPARED WITH THE DESIGN LOADS

MOD 0 OPERATIONAL CONFIGURATION	BLADE BENDING MOMENT (FT-LBS) AT STATION 40					
	FLAPWISE			INPLANE		
	PEAK TO PEAK	MEAN	CYCLIC	PEAK TO PEAK	MEAN	CYCLIC
Config. I	130,000	-65,000	+65,000	108,000	-18,000	±54,000
Config. II	70,000	-17,850	±35,000	102,850	-10,280	±51,420
Config. III	64,500	- 7,750	±32,250	80,000	-18,000	±40,000
Design	58,000	-23,400	±29,000	75,000	-11,200	±37,500

The supporting tests and correlations with analytical methods which lead to the understanding and solution of these engineering problems is the primary subject of reference 2. To facilitate a more detailed examination of the structural dynamics of the Mod-0 wind turbine system Appendix A has been included in reference 2 which provides basic Mod-0 geometry, mass and stiffness distributions, blade frequency spectra, and tower wake test results by NASA LeRC and Lockheed, also see references 3, 4 and 5.

The blade loading measurements taken for configurations I, II and III, presented in Table 1, were obtained with power loadings into a resistive load. Mod-0 synchronous operation with emphasis on the power/drive train dynamics is also reported in reference 2.

#### COMPARISON OF BLADE LOADS DURING OPERATION IN CONFIGURATION I, II AND III

The average peak to peak, mean and cyclic bending moments experienced during operation in configuration II were smaller than those measured during operation in configuration I. The results indicate that removal of the tower stairs and rails has a pronounced effect in reducing the flapwise bending moments in the rotor blades but little effect in reducing the inplane bending moments.

In comparing inplane blade bending moments for configuration II and III, the following observations are noted:

1. The average cyclic inplane bending moment was reduced by 22 percent during operation in configuration III.
2. The mean inplane bending moment increased by 75 percent during operation in configuration III.

The average peak to peak, mean and cyclic flapwise bending moments experienced during operation in configuration III are smaller than the loads measured during operation in configuration II.

The results indicate that by increasing the yaw stiffness of the structure between the nacelle and tower, the cyclic inplane blade bending moments can be significantly reduced. However, configuration III had little effect in reducing the cyclic flapwise blade bending moments, when compared to the blade loads encountered during operation in configuration II.

#### CONCLUSIONS AND RECOMMENDATIONS BASED ON MOD-0 TEST/ANALYSIS

- Accurate prediction of blade loads, particularly at higher frequency, requires an accurate description of tower, nacelle, and drive train dynamics.
- Blade high frequency mode tuning can be significantly affected by the structural dynamics of the tower and nacelle. Chord bending must be monitored if nacelle yaw system characteristics are changed to assure that blade loads don't become excessive.
- Good agreement between calculated and measured loads was obtained in an analysis that included only two blade dynamic modes, first flap and first inplane, and a blade quasi-steady torsion mode.
- Operation of the WTG, in its present configuration, should avoid power and rpm combinations that result in generator-armature resonance.
- To eliminate the armature resonance problem, use of either an increased stiffness coupling or an across-coupler damper appears feasible. Further analyses would aid in making a choice and are required to ensure an adequate design.

#### MOD-OA DESIGN CRITERIA AND LOADS

Strength and stiffness criteria were developed by NASA. Although the adequacy of these criteria to define ultimate strength has been shown to some degree during operation of the Mod-0 wind generator, the Mod-OA is located at

a different site, in a different environment, probably will be operated by personnel who have no experience with the Mod-O unit, and is intended to be operated for long periods of time, continuously integrated into an electrical power supply (varying loading). The Mod-OA unit will test the fatigue endurance of a wind turbine of this type long before comparable data are accumulated on the Mod-O unit. It is expected, however, that ultimate strength and stiffness criteria as applied to the Mod-OA unit are probably adequate. The major concern is the assurance of longevity under the total operating environment.

Differences in structural design between the Mod-O and Mod-OA wind turbine blades were simply the result of recommendations made by Lockheed during an analytical appraisal of Mod-O operational load measurements in the interest of increasing longevity and the results of recommendations of a NASA safety group. In the interest of expediency, all recommendations were not followed in the final design of the Mod-OA blade structure. For example, the skin thickness was reduced to comply with an existing manufacturing capability, and a cable-type blade retention system (which has been recommended by the NASA safety group) was deleted for expediency of cost/schedule.

The structural integrity of the Mod-O wind turbine blades is based on the four loading cases stipulated by Contract No. NAS3-19235. Airload distributions for each case were specified. However, it is conceivable that structural design criteria do not properly account for terrain or climatic differences from the Plumbrook site (site-specific criteria) nor, large fluctuations of loads which might occur as the result of being electrically tied into a power system different from the one the Mod-O unit has been tied to.

Design of Mod-OA blades was based on loads distributions for the four specified cases plus the upgrading recommendations. Recommended changes which were directed at the Mod-O design, assumed 50,000 hours of use at 40 rpm generating 100 kW in a 26 mph wind and that flow through the tower can be completely blocked (e.g., after a severe ice storm). These recommendations were used to aid the design of the Mod-OA blades. New design loads were calculated for the four cases and were adjusted to reflect data which was measured on the Mod-O unit.

## MOD-OA FATIGUE

Fatigue life prediction entails many uncertainties. For example there are many situations that might cause the blade loads spectrum to exceed that which was established for design; and untested structural details can cause concern regarding blade life prediction even under known load conditions.

The curve of figure 2 is the result of a fatigue analysis, reference 7, made for the Mod-OA blade structure. The analysis assumed:

- Blade station 637.5 is probably most critical.
- Airload distributions which had been specified for the Mod-O blades apply to the Mod-OA.
- Allowable stresses have a 99% probability of occurring.
- The wind turbine will operate at speeds up to the cut-off wind speed for 50% of the time.
- The quality of structure (design and manufacture) is comparable to that of an airplane wing; i.e., stress concentrations exist at some local structural details.

The prediction precludes effects of fretting, corrosion, or other unpredictable damage. The figure shows that a life of 30 years is attainable with a cut-off wind speed of 41 mph (initially specified as 26 mph). However, even when a high quality is sought in a development unit it is possible that a lesser quality will exist at some local details.

Results of the fatigue analysis were also plotted (figure 3) to show what might be expected if:

- loads or stresses are different from those calculated.
- the quality of the structure is different from expected.

Interpretation of figure 3 introduces the need to define two symbols which are common to fatigue analysis,  $S/S_o$  and  $K_T$ .  $S$  represents stress, and the subscript,  $o$ , merely signifies "original"; so  $S/S_o = 1.0$  represents the values used in the analysis whereas a ratio above 1.0 would indicate that actual stresses will be greater than those used in the analysis. The  $K_T$  value

is a measure of the quality of the structure; it reflects holes, scratches, cut-outs, etc. A value of  $K_T = 4.5$  is representative of the quality of aircraft structure generally sought during design, but as implied earlier, it's not unusual for  $K_T$  values to be as high as 6 or 7 at some structural details.

It's noteworthy from both figures (2 and 3) that a change of a few mph of cut-off wind speed, or a small change in either  $K_T$  or  $S/S_0$ , can mean a change of very many years of life. Had cost/schedule permitted, tests of structural details could have significantly reduced the uncertainty of fatigue life prediction.

### INSPECTIONS

It must be realistically assumed that cracks, corrosion, and fretting will appear during the operation of any new engineering equipment. Even in aircraft, where structural analyses are usually backed by many tests, the most damaging fatigue loading on an aircraft wing structure occurs during the takeoff-fly-land cycle, which on a high-time commercial airplane might occur around 100,000 times. A helicopter rotor blade has a finite replacement life of 2000 hours and is generally subjected to load cycles of the order of 20,000 per hour. The wind turbine blades, which are expected to operate for 30 years, will experience millions of cycles (approximately 12 million cycles/year).

The high number of load cycles per unit time is of concern with respect to rates of crack propagation. If a small crack exists due to a material flaw or a seemingly inconsequential scratch or crack, the high frequency of load reversals can cause the crack to grow rapidly to a critical size. To circumvent this possibility most of the structure was designed to be fail safe, i.e., consisting of multiple elements. However, fail safety requires that growing damage be found before it grows to a critical length. Thus, regular inspections are necessary.

The flange root tubular steel fitting and the hub structure are monolithic (not fail safe).



The wind turbine is expected to be in service (operating) about 50 percent of the time, and will be out of service (but possibly in a violent environment) the rest of the time.

Fretting is known to accompany a high frequency of load reversals, so appropriate inspections must be established to permit early detection of cracks due to fretting.

The installation is subject to exposure to weather which can become a corrosion problem. The recommended inspection interval must also take this phenomenon into account.

#### INSPECTION PERIODS

Start inspections after the first 500 hours of operation. All inspections which are recommended for the root area and basic blade should be performed every 500 hours of operation ( $10^6$  cycles). After five inspections without cracks, severe corrosion, or fretting, increase the inspection period to 1000 hours. After five inspections beyond the 1000 hour period, a further extension should be considered. A careful review of the loads experience, inspection reports, and other pertinent results should be made before extending the inspection period.

X-ray the entire blade once per year.

It is important to derive similar inspection periods for hardware other than the blade.

It is difficult to establish an inspection interval for fretting and corrosion because there exists no experience with wind turbines on which to base such an interval; except the NASA experimental operations with the Mod-0 wind turbine. It would be expected that the inspection interval will increase as operating experience accrues.



#### MOD-OA 200 kW OPERATIONAL EXPERIENCE

The MOD-OA wind turbine entered service January 1978 in Clayton, N.M., following several years of development by NASA-Lewis on a similar machine, the MOD-O (of the same external geometry as the MOD-OA), at Plum Brook, Ohio.

Operating loads were monitored continuously during the first three months of operation. There was no indication that operating limits would ever be exceeded, so the monitoring operation was relaxed. The new monitoring procedure was to record and erase at 45-minute intervals (of operation) and to examine operating loads occasionally.

At some time following initial checkout, structural damage did accrue. An automatic shutdown device did not prevent high loads. The first sign of difficulty was a discoloration of some fasteners; rivets began to loosen; and a crack in the skin was found. Because excessive loads were not at first apparent, the initial assumption was that faulty workmanship during blade manufacture caused the structural damage. However, reexamination of some records showed that loads in excess of operating limits were encountered. They occurred during nacelle yawing and, according to NASA sources, the total time spent in nacelle yawing during the lightly monitored 3-month period was 58 hours, reference 12.

These excessive loads could have caused the damage; however, it's probable that other loads, as high or even higher, were encountered at some time when no record was made. This documented situation indicates that the design of the blade structure is very forgiving; even though operating limits were exceeded, damage was minimal and repairable.

Prior experience on the MOD-O and subsequent investigations of the MOD-OA system strongly suggest that these loadings were caused by massive yaw stiffness degradation that led to a 2P resonance of the nacelle/tower system in the yaw axis. This is thought to have occurred only during nacelle yawing operation.

#### MOD-OA MEASURED BLADE LOADS

Results of an analysis of the data of reference 8 (NASA PIR 44), the

variation of cyclic and mean flapwise and chordwise bending moment at Blade Sta 40 with wind speed, are superimposed on the graphs of reference 9 (NASA PIR 58) in figures 4, 5, 6, and 7. There is approximate agreement of most of the data, especially mean bending moments, where the data of PIR 58 are believed to be for normal operation. The cyclic flapping bending moments, however, are significantly larger and the chordwise loads somewhat larger than reached earlier. (Data are from tests at Clayton, New Mexico.)

Lockheed's calculated blade loads are shown in figures 4, 5, 6, and 7. These were calculated for steady winds with zero yaw. The steady wind loads represent average loads ( $0\sigma$ ) presuming that positive and negative gusts and yaw errors do not affect the average. These loads were used for structural evaluation of the MOD-0A wind turbine blades.

$3\sigma$  curves were predicted by assuming  $30^\circ$  yaw error and gusts. With mean and  $3\sigma$  curves available, the  $2\sigma$  curve is approximated for a Gaussian distribution and compared with measured data for high winds (PIR 44). The low wind speed  $2\sigma$  curve is obtained, similarly, by extrapolating the data of PIR 58 from  $0\sigma$  (mean) and  $1\sigma$  to  $2\sigma$ .

An interesting note is that the variations of flapwise and chordwise loads that were calculated based on a stiff yaw system were slightly conservative when compared with the loads recorded during the high winds of January 7, 1978 (PIR 44). This does not necessarily mean that safe loads were not exceeded, however, since the variations shown above were used for blade evaluation only to the cutoff wind speed (40 mph). Loads due to higher wind speeds were not considered in the fatigue spectrum since no such loadings (in routine service) were expected, reference 11.

The  $2\sigma$  loads are shown for comparative purposes only. In defining the fatigue load spectrum, all levels of loads must be considered.

#### LOADS DUE TO DEGRADED YAW SYSTEM OPERATION

The magnitudes of the loads measured indicate that the system suffered deterioration prior to May 18, 1978. Furthermore, the postulation that at least part of this behavior was due to nacelle yaw stiffness deterioration

appears to be correct by reference to the wave form of the inplane Blade Sta 40 bending moment history shown in figure 8. It displays a dominant 3P harmonic that has been seen, in the analyses of reference 10, only with a tower of low torsional and lateral bending stiffness.

The chordwise and flapwise bending moment histories calculated for Case 2 of reference 10 are also shown in figure 8. Aside from chordwise oscillation amplitudes being about half those of the Clayton data of 5/18/78, they agree very closely in wave form. The flapping bending moment also shows good agreement, even in magnitude, with that measured.

In Case 2 of reference 10, a MOD-0 tower with an arbitrarily low stiffness in lateral bending and torsion was employed. A review of the data showed tower resonance at about 1.4 Hz or approximately at 2P. The 3P blade inplane bending oscillation appears to be due to the tower 2P resonance. In the case of the MOD-OA system, this would roughly be equivalent to operating with yaw drive stiffness somewhat less than occurs with a single-drive unit (or approximately a 90 percent stiffness reduction). But here the agreement between the two situations ends. In Case 2, the blade inplane cantilever frequency was 3.80P (in MOD-OA it is supposed to be 4.80P) the wind speed was 40 mph (instead of the reported 25 mph) and the tower shadow employed in Case 2 was that with the stairs in (instead of the much lower MOD-OA tower shadow). The power output at the shaft was 133 kW instead of the 242 kW of MOD-OA.

Since the measured flapping loads would agree much better with calculated loads at a higher wind speed, the wind speed during the loads measurement may be in error.

#### STRUCTURAL REPAIR OF CLAYTON, N.M. BLADES (004 AND 005) AND SERIAL UPGRADE OF MOD-OA PRODUCTION BLADES

Mod-OA blades 004 and 005 were taken to NASA Lewis Research Center, Cleveland, Ohio, in June 1978 after approximately 1200 hours of service at Clayton, New Mexico. The complete blade was inspected for structural defects using nondestructive testing methods. The steel root fitting was removed and the root end of the blade inspected further. The damage found (after 2.8 million rotations) was localized between stations 48 and 125. The blade span is

750 inches with the first rib located at Sta 48. This work was supported by Lockheed both at Clayton, N.M. and Cleveland, Ohio. Following inspection, preliminary designs were made of reinforcement doublers for repair of the distressed areas. Fretting of the rib interface at Sta 48 was also found and various conceptual designs were examined by Lockheed and NASA engineers, but resolution of this subject presently is dependent on development testing in the NASA laboratories.

The NASA modifications were installed at Cleveland on blades 004 and 005. Similar designs are documented on Lockheed Drawings 1900031 and 1900032 to provide Lockheed Aircraft Service Company, Ontario, California, the means to incorporate these modifications into the current production blades 008 and 009. The root details at the rib station 48 will be supplied by NASA Lewis Research Center directly to LAS upon completion of their laboratory test program. It is expected that the additional blades 010 and 011 will be similarly modified during assembly and that blades 006 and 007 will eventually be retrofitted to this same configuration.

The rationale for the incorporation of the repair modifications into production blades is that hot spots have been identified by the abnormally high loadings. These loadings can be considered as an accelerated fatigue test that has pointed out areas in which increased loads margins can provide added protection.

Since blades 004 and 005 structural repairs were made on an inspect-and-repair-as-necessary basis rather than through an ongoing analysis/design effort, it is recommended that the operational limitations recommended in LR 28395, 10 January 1978, for continuous loads monitoring and frequency of inspections be followed.

Other MOD-OA blades that are modified in the same manner as 004 and 005 should also adhere to the operating limitations recommended in LR 28395.

#### OPERATION OF MOD-O BLADE AT CLAYTON, N.M. DURING STRUCTURAL REPAIR OF MOD-OA BLADES

The MOD-O blades were installed on the Clayton, N.M. wind turbine in June 1978 as an interim means to continue operation of the system. The 200 kW capability of the wind turbine was retained even though these blades were designed for the 100 kW experimental unit and had been through the early development phases of wind tower shadow, emergency feather and nacelle/tower yaw stiffness. Following approximately 800 hours of this added service and return of the repaired MOD-OA blades these MOD-O blades were returned to Cleveland, Ohio for inspection and overhaul. The fretting at rib station 48 was at this time quite measurable by feeler gage. The shell structure, as in the case of the MOD-OA blades outboard of station 100 to the tip station 750, was in excellent condition. Upon removal of the steel root-end fitting it was found that the rib at station 81 was cracked in several locations and that the safety attachment between the steel tube and the blade web had some elongated holes. This indicates that centrifugal loads were being partially carried by this backup load path. Although the fretting might be cured by elastomeric bearing or aluminum bronze bushing applications, it seemed the rib replacement repairs at blade station 31 warranted a more direct method consistent with tooling requirements. Therefore, the repair method proposed addresses the repair, fretting and tooling as one problem. The proposed modification to the MOD-O blades as shown on figure 9 and figure 10 has been adapted and will be implemented in the near future.

#### CONCLUSIONS

The MOD-OA wind turbine which entered service in January 1978 in Clayton, N.M. was found to be damaged locally between stations 48 and 125 after 2.8 million rotations. Loads due to degraded yaw stiffness and fretting at rib station 48 were the factors which have been identified as primary to this distress. The repaired blades have now operated an additional 2000 hours (4.8 million rotations) without further problem.

It is noteworthy that the fatigue analysis predicts that station 637.5 is the most critical and these sections have 7.6 million rotation cycles.

The shell structure of the MOD-OA and MOD-O blades outboard of station 125 are both in excellent condition. The latter unit now has a total of 1300 hours (3.1 million rotations).

The correlation between test and analysis has been good and currently no unexplained problem areas exist.

Refinement of the emergency or safety shutdown feathering rates and procedures are desirable to minimize the large load transients which can occur.

Since this aluminum blade structure has been shown to be forgiving, the primary focus of additional safety features should be the monolithic hub which might lend itself to the cable-type retention previously mentioned as a NASA safety group consideration.

#### REFERENCES

1. Puthoff, Richard L.: Fabrication and Assembly of the ERDA/NASA 100 Kilowatt Experimental Wind Turbine. NASA TM X-3390, 1976.
2. Linscott, Bradford S.; Glasgow, John; Anderson, William D, and Donham, Robert E.: Experimental Data and Theoretical Analysis of an Operating 100 kW Wind Turbine, NASA TM-73883, 1978.
3. Glasgow, John C.; and Linscott, Bradford S.: Early Operation Experience on the ERDA/NASA 100 kW Wind Turbine - Rotor Blade Loads. NASA TM X-71601, 1976.
4. Cherrit, A.W.; and Gaidelis, J. A.: A 100 kW Metal Wind Turbine Blade Basic Data, Loads and Stress Analysis. (LR-21753, Lockheed-California Co.; NASA Contract NAS3-19235.) NASA CR-134956, 1975.
5. Donham, R. E.; Schmidt, J.; and Linscott, B. S.: 100-kW Hingeless Metal Wind Turbine Blade Design, Analysis and Fabrication. Presented at the 31st. Annual National Forum, American Helicopter Society, (Washington, D.C.), May 13-15, 1975.
6. Cardinale, S.V.; Anderson, W.D.; Gaidelis, J.A.; Rhodes, J.E.: MOD-O Wind Turbine - Task III - Correlation of Analytical and Actual Loads Data, Task IV - Potential Structural Blade Modification, Lockheed-California Company, LR 27780, August 20, 1976.
7. Cardinale, S.V.: MOD OA Wind Turbine Blades Operating Limitations - Executive Summary. Lockheed-California Company, LR 28395, January 10, 1978.
8. Shaltona, R.: MOD OA-1 Data for January 7, 1978 High Gusty Wind Conditions. PIR No. 44, 2/3/78.



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9. Richards, T.; Newstadter, H.; Spera, D.A.: MOD OA Blade Loads. PIR No. 58, May 10, 1978.
10. Watts, G.: Results of Analyses performed under Contract NAS3-20593 Structural Dynamic Analyses of the MOD-OA Wind Turbine Blades and Drive Train. Completed Sept. 1977. (A total of 36 cases).
11. Anon.: Clayton Wind Exceedance Curve for one Year at 30 ft Elevation - One Sheet.
12. Watts, G.: Evaluation of Measured Blade Data MOD-OA Wind Turbine. Lockheed-California Company, LR 28706, August 15, 1978.

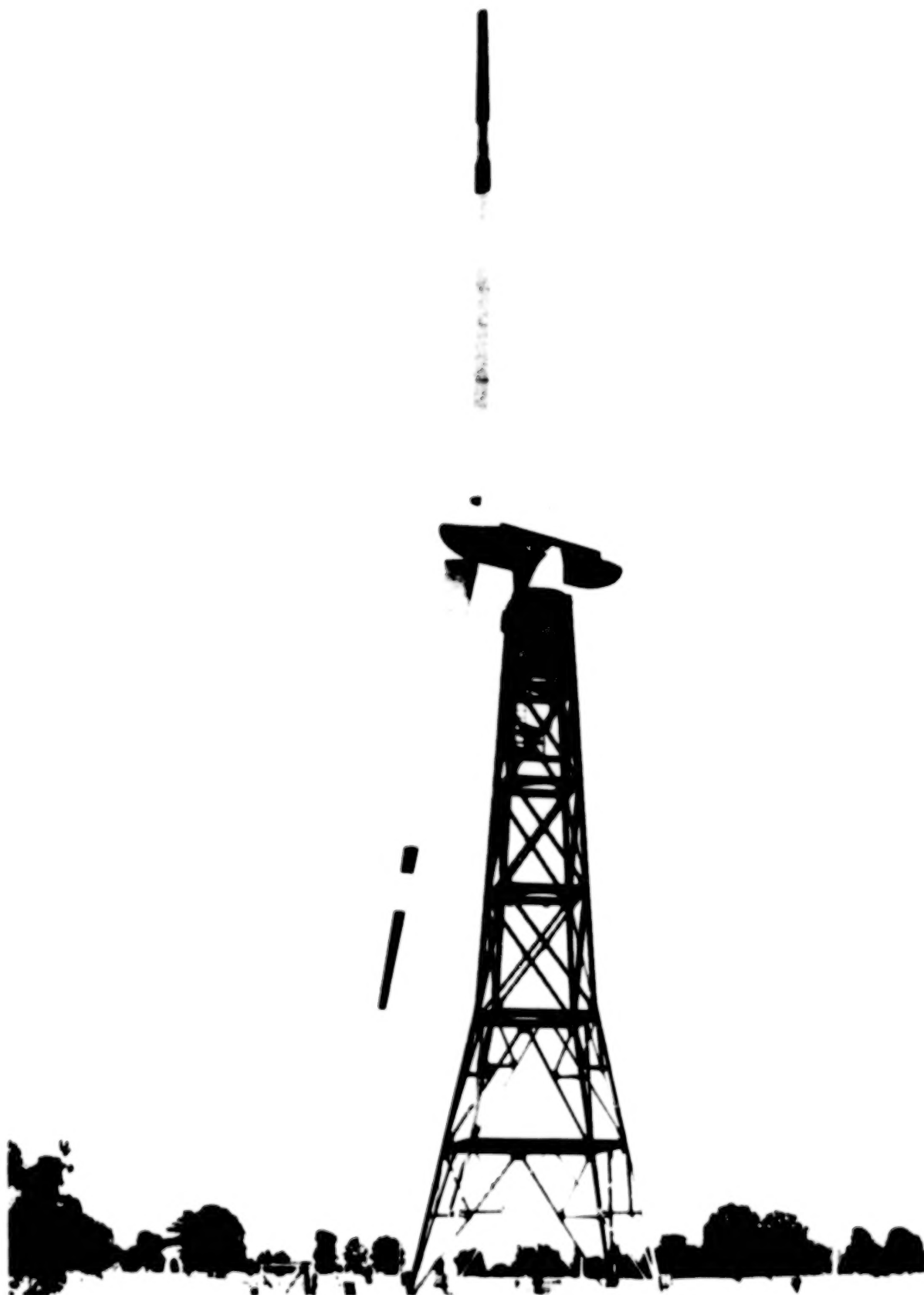


Figure 1. Mod-O Wind Turbine - Current Configuration (Mod-O and Mod-OA)  
Without Stairs or Rails.

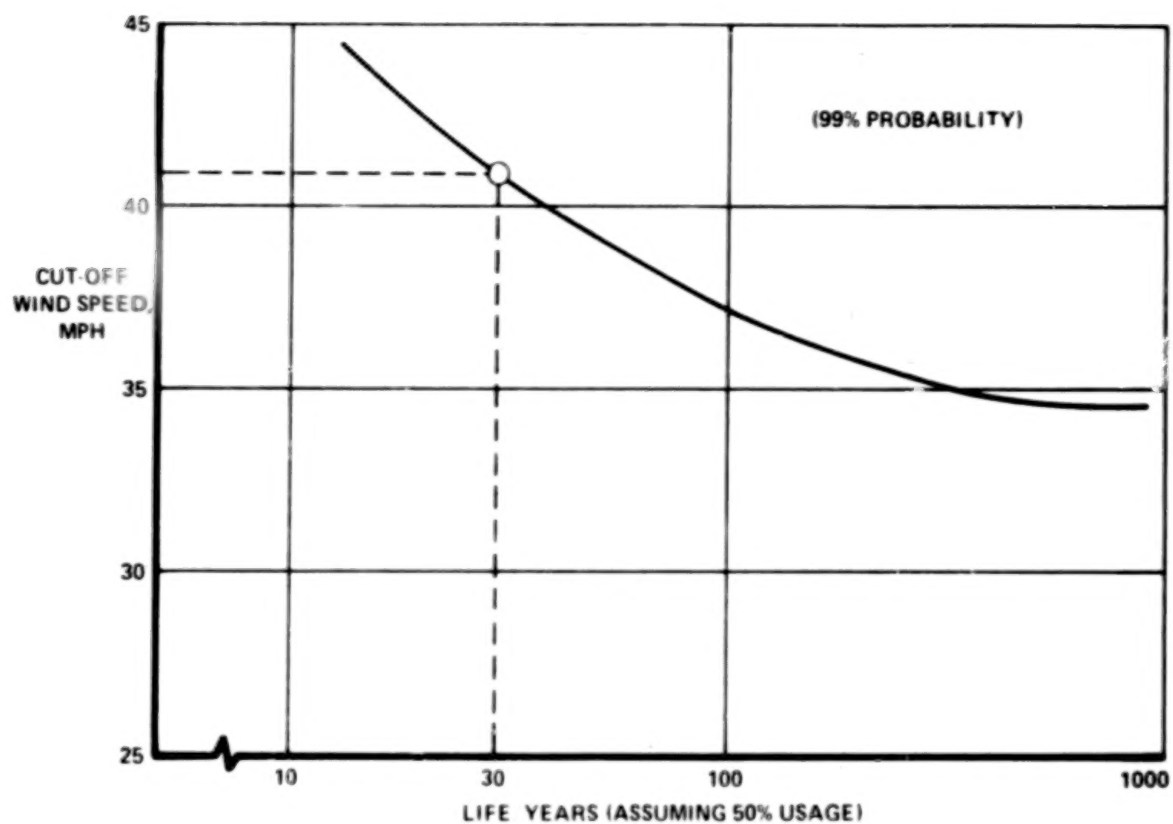


Figure 2. Result of Fatigue Analysis of Mod-OA Blade (at Station 637.5) Assuming Structure Quality Comparable to that of a Typical Airplane Wing Structure.

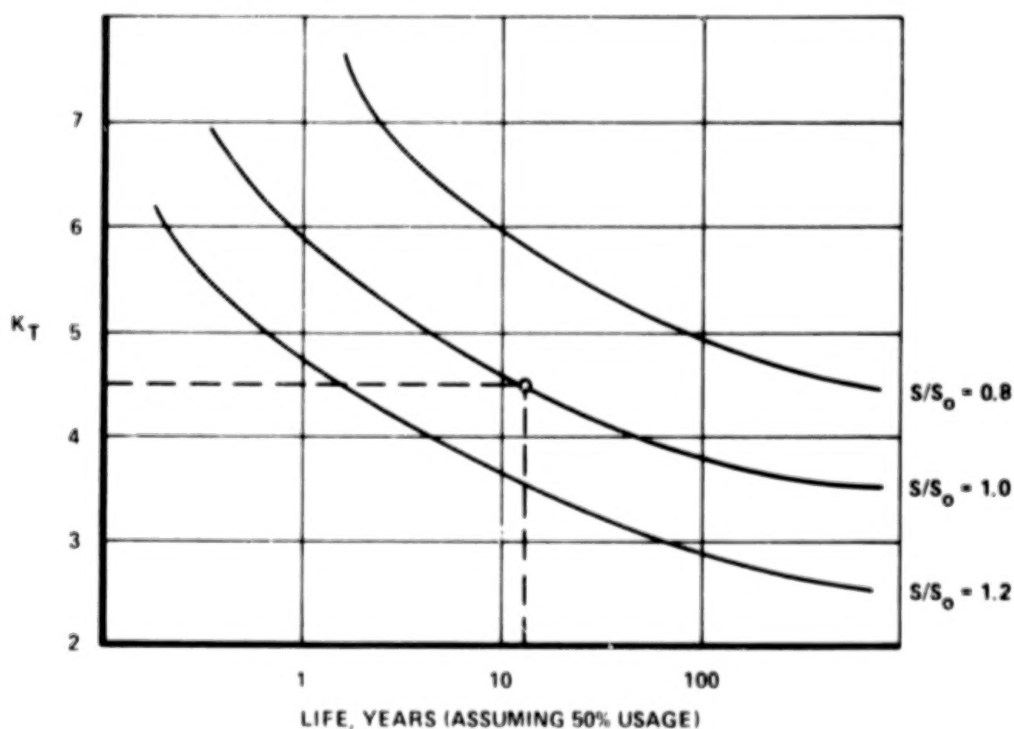


Figure 3. Result of Fatigue Analysis of Mod-OA Blade (at Station 637.5) Indicating What Might Happen to Life as  $S/S_0$  and  $K_T$  Vary.

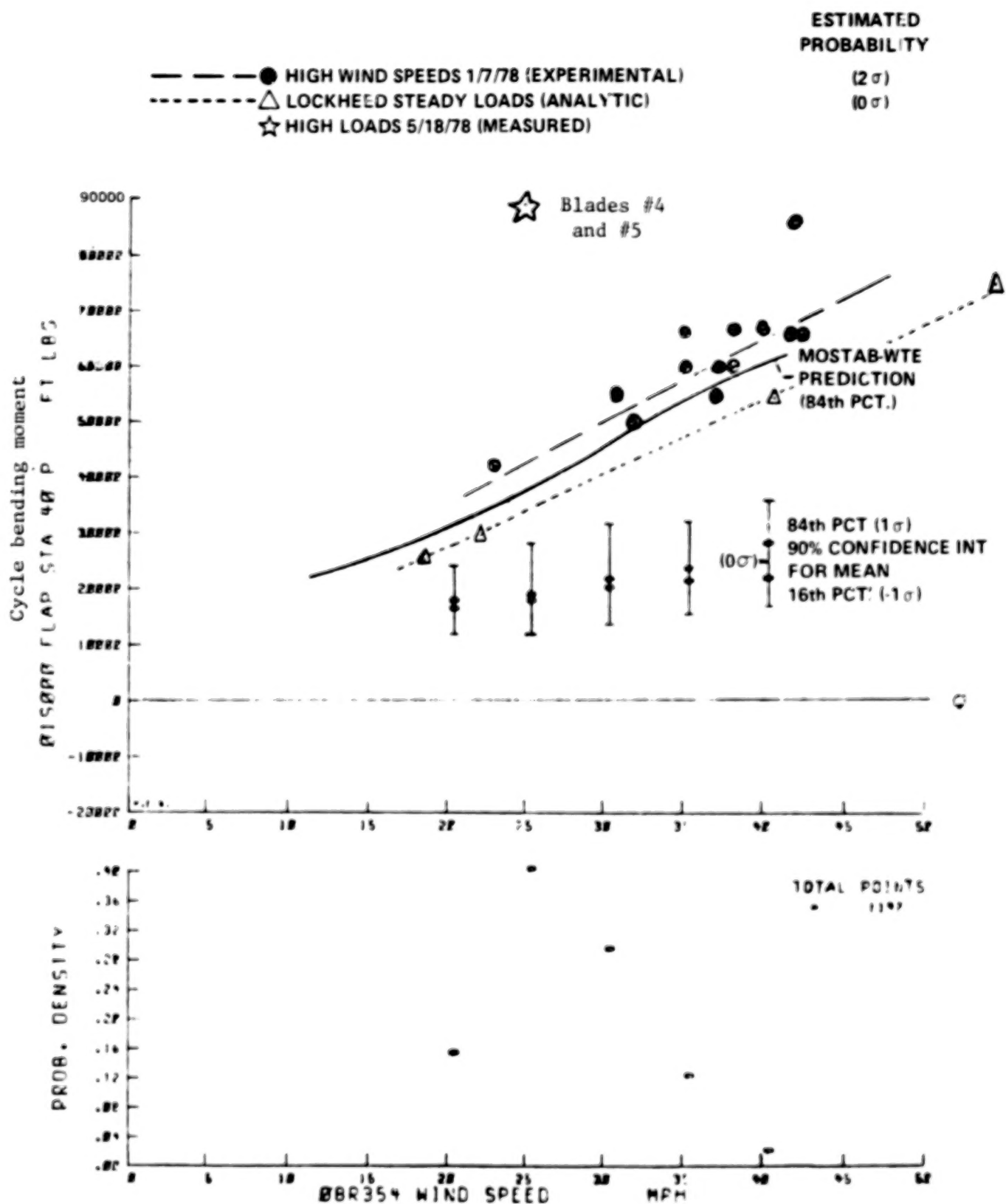


Figure 4. Cyclic Flapwise Moment at Sta 40.

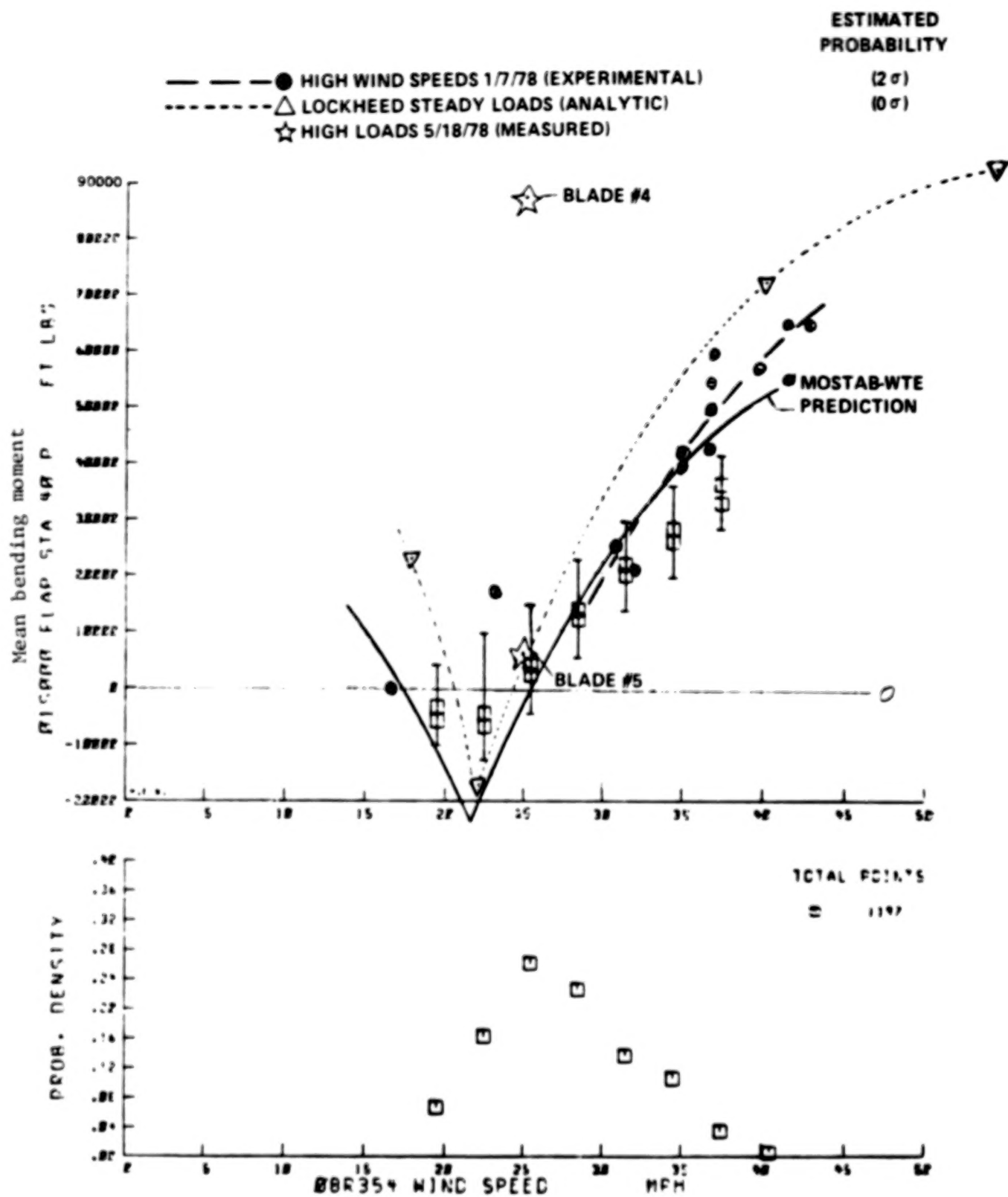


Figure 5. Mean Flapwise Bending at Sta 40.

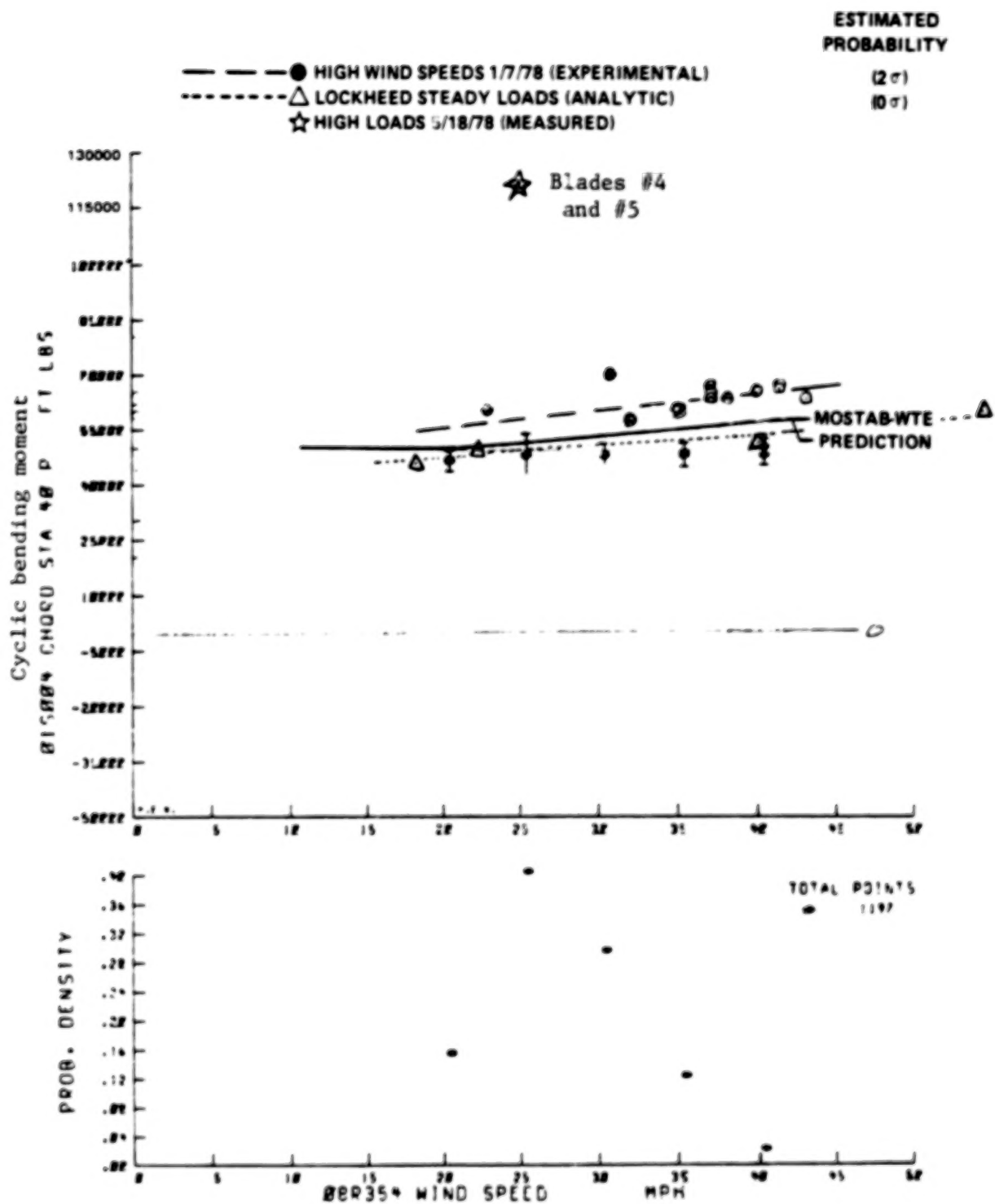


Figure 6. Cyclic Chordwise moment at Sta 40.

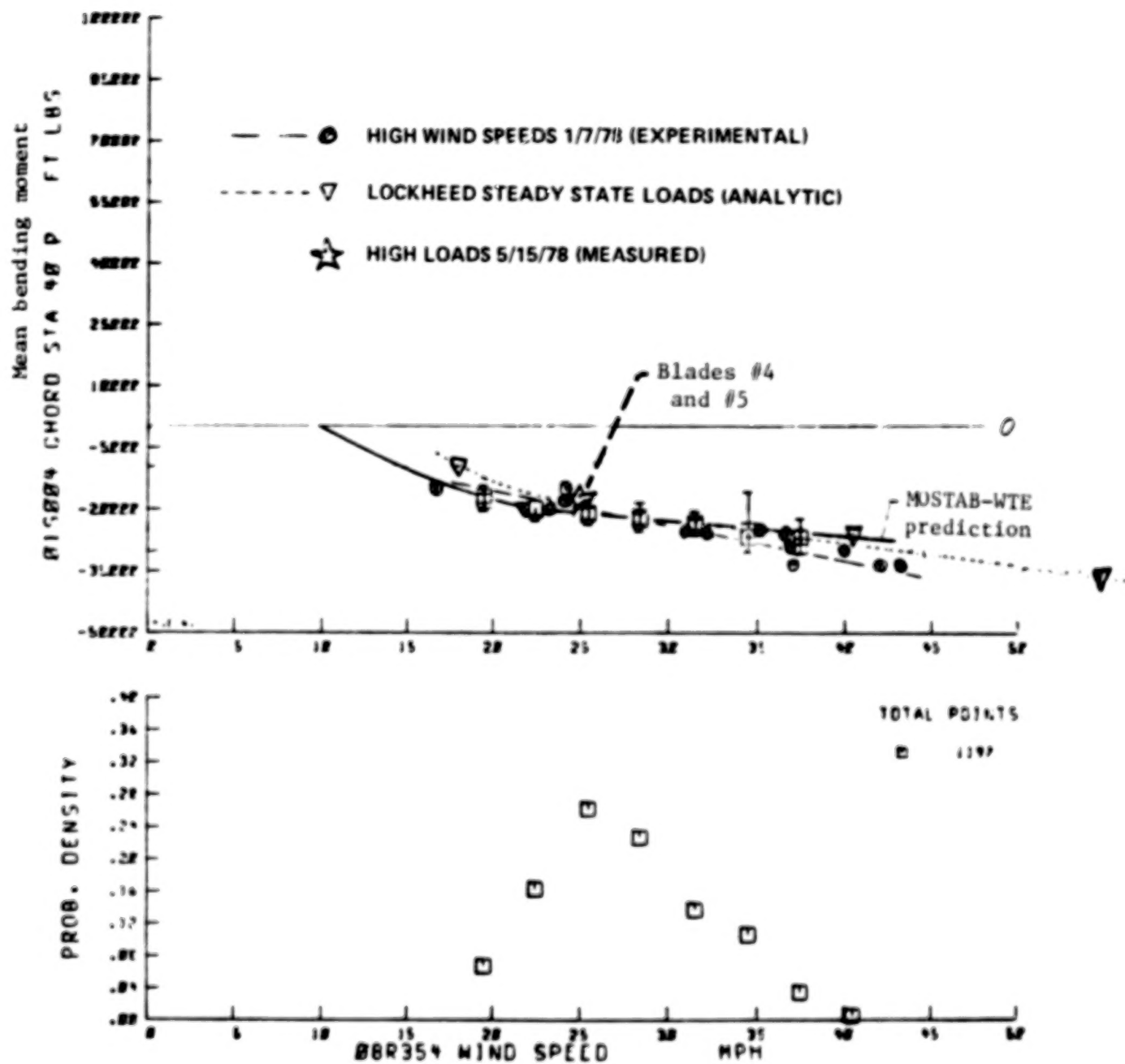


Figure 7. Mean Chordwise Bending at Sta 40.



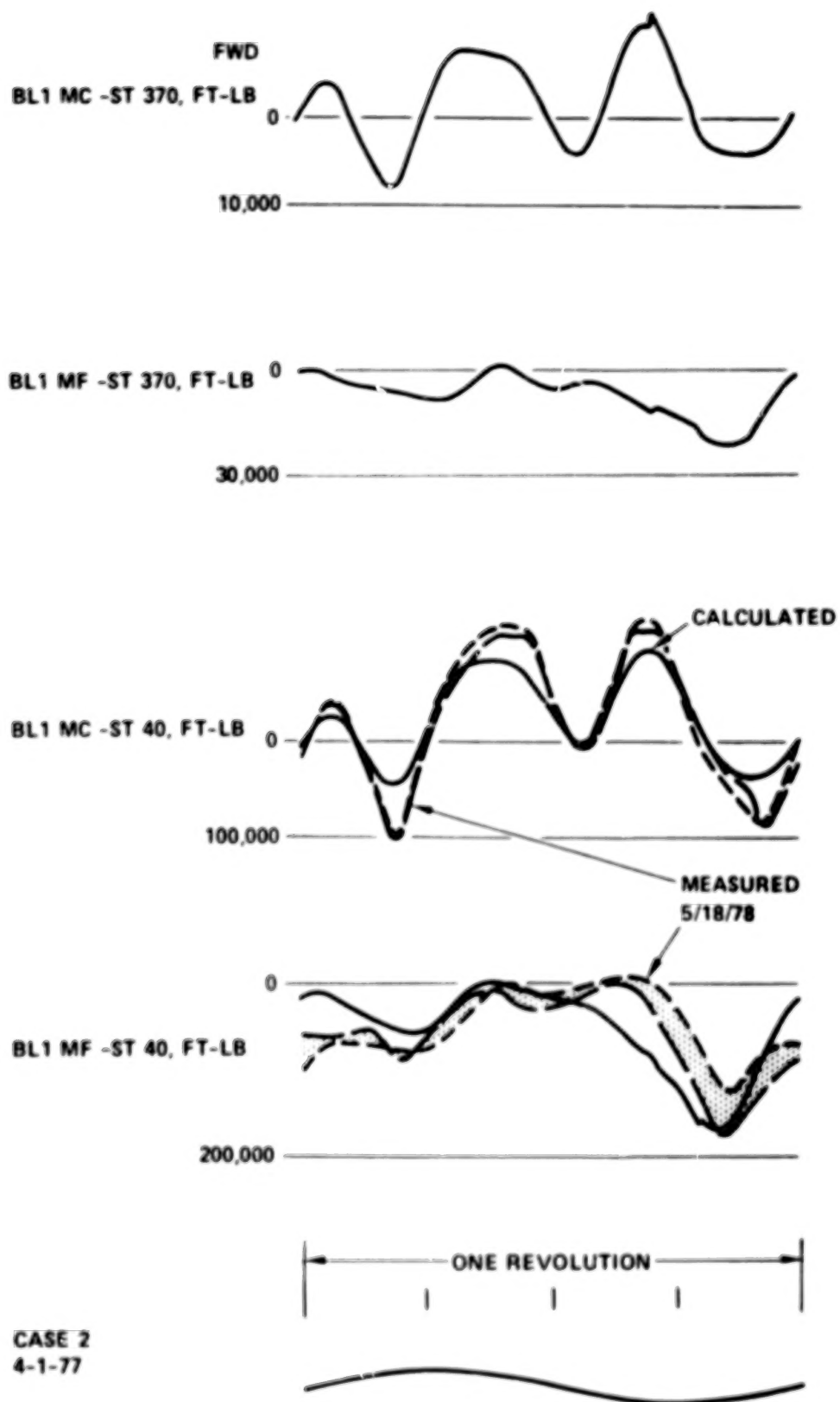


Figure 8. Calculated and Measured Blade Loads With Reduced Tower Stiffness.

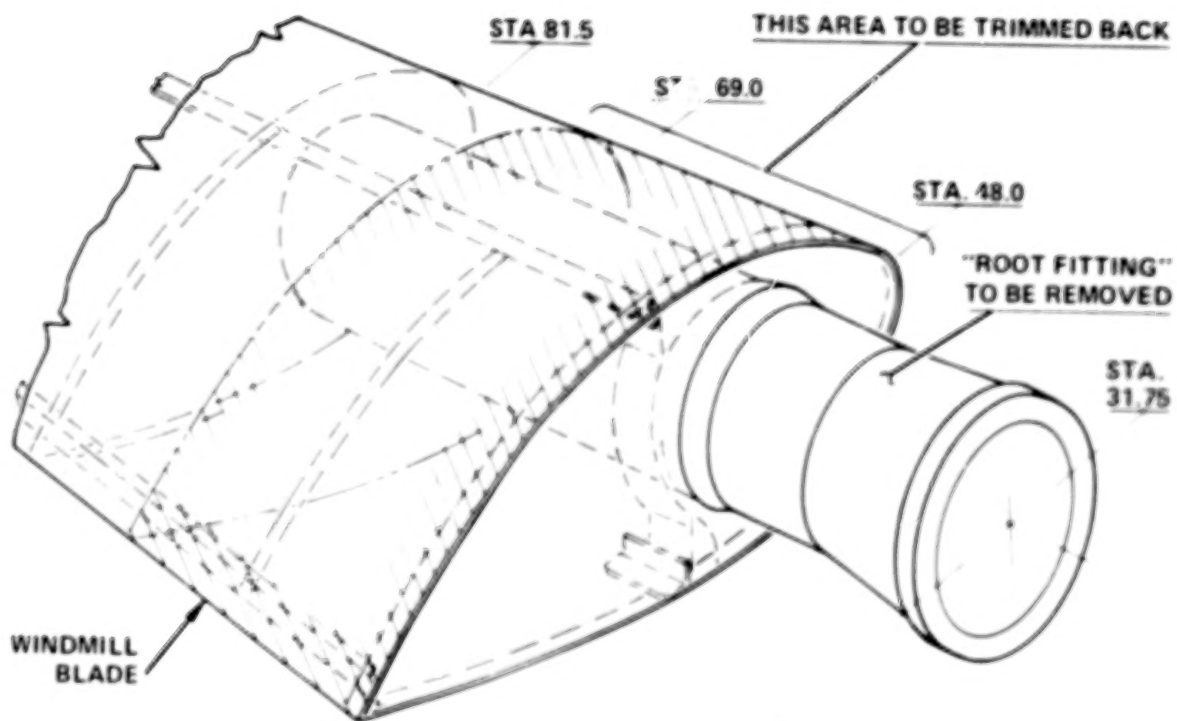


Figure 9. Blade Root End Before Modification (Existing).

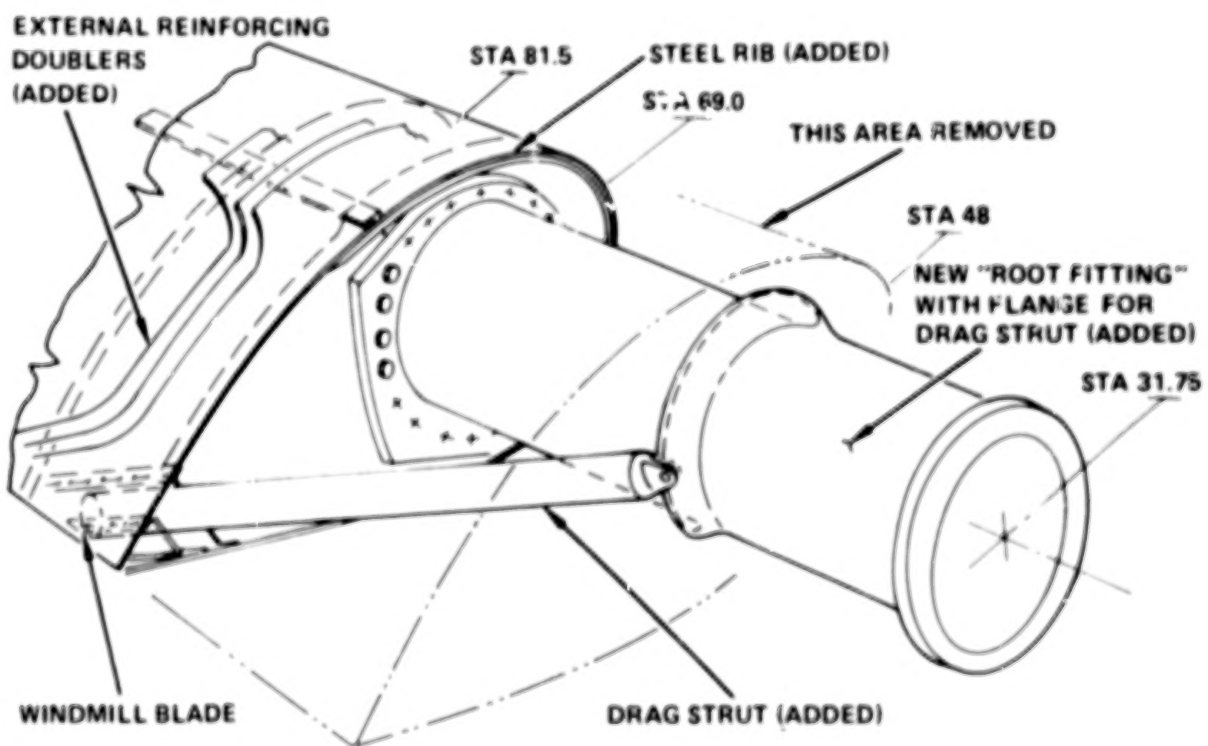


Figure 10. Blade Root End After Modification.

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## DESIGN, FABRICATION, AND TEST OF A STEEL SPAR WIND TURBINE BLADE

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### INTRODUCTION

One potential means for reducing the costs of wind turbine blades is to use a mass produced structure as the primary structural member of the blade. Tapered beams such as those used for utility poles are the type of mass produced structure envisaged. The airfoil shape could be formed by light-weight foam or light-weight ribs overwrapped with fiberglass cloth. In order to determine the feasibility of this concept, a 60 ft. steel spar blade was designed. Using this design, two blades were fabricated at the Lewis Research Center and tested on the Mod-0 wind turbine (ref. 1).

This paper describes the design and fabrication of the blades. Performance and blade load information is given and compared to analytical prediction. In addition, performance is compared to that of the original Mod-0 aluminum blades. Costs for building the two blades is given, and a projection is made for the cost in mass production. Finally, design improvements to reduce weight and improve fatigue life are suggested.

### BLADE DESIGN AND ANALYSIS

The purpose of this program was to show that a wind turbine blade based on a steel spar could be fabricated in a satisfactory manner. Once fabricated it was necessary to show that the blades performed adequately on Mod-0. A 30-year blade life was a secondary consideration in this program.

#### Design Concept

A schematic of the steel spar blade design is shown in figure 1. Figure 1(a) shows the steel spar. Because of the constraints placed on the design by the Mod-0 hub flange dimensions and the availability of steel spar material, the steel spar extended from station (sta.) 148 in. to sta. 750. The space between the hub flange (sta. 32) and the steel spar flange was filled with a high strength steel tubular extension section. To reduce the blade loads it was necessary to reduce the precone angle from the 7° built into the Mod-0 hub to 3°. The steel spar is at a 4° angle to the extension section. The steel spar is made in two sections which are welded together at sta. 558.

This was done to allow the use of lighter wall material on the outboard section of the blade.

The blade planform and a typical cross section are shown in figure 1. The leading edge airfoil shape is formed with foam while the trailing edge shape is formed with wooden ribs. The skin is fiberglass cloth. Details of the design are given in a subsequent section.

#### Design Specifications

Blade design requirements and additional rotor information are given in table I. For comparison purposes, the same information for the original Mod-0 aluminum blades is given in the table. The steel spar blade is a much simpler blade with no twist and a constant thickness ratio. The rotor operating speeds were chosen to optimize annual energy capture.

The spar design was based on a single load case. That was the 120 mph wind case which was taken to be equivalent to a uniform pressure of 50 lb./ft.<sup>2</sup> on the surface of the blade. Because of the short term nature of the operation of this blade on Mod-0, no fatigue load cases were required.

#### Design Allowables

The critical area of this design is the spar to flange weld area. Initially it was assumed that the flange material would have a yield strength of 60,000 psi. However, the actual material had a yield strength of only half this. Table II gives the measured material properties for the various spar components. The allowable strength was taken as the minimum measured strength.

Because of the reduced allowable in the flange material, additional analysis was done with the high wind load case. In addition, a rotor overspeed to 48 rpm load case was investigated. The high wind case showed that the spar flange yield strength allowable could be exceeded in winds as low as 87 mph. However, the stress was well below the ultimate strength in winds up to 120 mph. The overspeed case also showed the yield allowable was exceeded but again the stress was well below ultimate.

#### Predicted Fatigue Life

The cyclic blade loads associated with operation at the cut out wind speed (40 mph) were used in predicting a fatigue life for these blades. These loads would provide a conservative estimate of fatigue life. The MOSTAB-WTE code (ref. 2) was used to calculate spar loads and stresses. Cyclic and maximum spar stresses are plotted in figure 2 for operation at 40 mph wind speed. The critical portion of the blade is the spar to flange weld at sta. 153. Using the stress amplitude at this station, both the Structural Welding Code and the AISC Code were used to estimate fatigue life. Fatigue life predictions are shown in figure 3. The Structural Welding Code predicted a life of  $2.9 \times 10^5$  cycles, while the AISC Code predicted a life of  $6 \times 10^5$  cycles. Based on these predictions, it was determined that the blades could be

safely operated for 100 hours (about  $2 \times 10^5$  cycles) before reinspection of the welds is required.

#### BLADE FABRICATION PROCEDURE

The first step in the blade fabrication procedure was preparation of the steel spar. The spar-to-spar and spar-to-flange welds were made. The welds were inspected by dye penetrant and x-ray. The outer surface of the completed spar was then sandblasted and coated with an epoxy resin.

The remaining steps in the fabrication procedure are illustrated in figure 4. The wood leading edge ribs, spaced 12 in. apart, were bonded into place. A fiberglass tube for holding balance weights was inserted through holes in the ribs and bonded into place. Foam pieces, previously cut to shape, were inserted between the ribs and bonded to the spar. This assembly was then overwrapped with three layers of fiberglass cloth/epoxy and cured.

Next, the wood trailing edge ribs were bonded into place, again at 12 in. intervals. A wood trailing edge piece was bonded to these ribs. To provide additional support for the ribs, pieces of foam overwrapped with fiberglass were inserted between the ribs and bonded to them.

The last major process in the fabrication procedure was the installation of Razorback cloth. Razorback is a specially treated fiberglass cloth which shrinks when a cellulose acetate butyrate (CAB) dope is applied. This material is used in general aviation and provides a very strong and smooth surface. The assembly shown in figure 4 (e) was overwrapped with two layers of Razorback and doped with CAB.

The final step in the procedure was painting. Photographs taken during blade construction are shown in figure 5.

#### BLADE TESTS

The steel spar blades were mounted on the Mod-O hub in late September of 1978. They were removed from the hub in late February of 1979. During that time they operated for about 75 hours during a variety of tests. After dismount the blades were inspected. The Razorback skins showed no sign of deterioration and dye penetrant check of the spar-to-flange weld revealed no cracks. This section presents blade weight, balance and natural frequency information obtained before the blades were operated, and blade performance and loads during operation on Mod-O.

##### Weight, Balance and Natural Frequency

Blade weight and balance data are summarized in table III. As fabricated, the weights of the two blades were within 28 lbs. (approximately 1 percent) of each other. The total weight was 3617 lbs. compared to 2000 lbs. for the Mod-O aluminum blades.

For analysis purposes the blade natural frequency and mode shape were calculated using a finite element model. This model considered the mass and stiffness of the steel spar and extension piece, but only the mass of the wood, foam and fiberglass. The frequency and mode shape obtained from the model were used in the MOSTAB-WT (ref. 2) code for predicting blade loads.

The first flatwise and edgewise cantilever bending frequency predicted by the model was 1.88 Hz. The actual blade frequencies were obtained with the blades mounted on the Mod-0 hub. The first flatwise frequency was measured at about 1.75 Hz. and the first edgewise was measured at about 1.85 Hz. The small difference between the flatwise and edgewise frequencies indicated that the wood and fiberglass contributed only slightly to the blade stiffness.

#### Blade Performance and Loads

During the tests of the steel spar blades, Mod-0 was operated with the tower in two distinct modes. The first was the hard tower mode where the tower had a first bending frequency of about 2 Hz.; the second was the soft tower mode where the tower had a first bending frequency of about 0.8 Hz. This frequency change was achieved using a fixture that was placed between the Mod-0 tower and its foundation (ref. 3). The measured performance and loads described in this section were for both the hard and soft towers. In general, the tower natural frequency had very little effect on performance and blade loads.

The measured blade performance is compared to that predicted by the PROP code (ref. 4) in figure 6. The blades performed slightly better than predicted. Their performance is compared to that of the Mod-0 aluminum blades in figure 7. The performance is nearly equivalent which was unexpected because of the steel spar blades having a larger root cutout and no twist. These aspects detrimental to performance must have been offset by the better surface smoothness achieved with the type of fabrication used for the steel spar blades and by their larger chord length.

A comparison of measured and predicted flatwise steady and cyclic blade root loads is shown in figure 8. The predicted loads were obtained from the MOSTAB-WT code (ref. 2). There is good agreement with the cyclic loads. At the higher wind speeds the code overestimates the wind shear resulting in higher predicted cyclic loads. The difference between the predicted and measured steady component of the flatwise loads is probably due to some error in the mass distribution of the model used in the code. In addition, the actual rotor speed was slightly less than that used in the code.

Measured and predicted chordwise steady and cyclic blade root loads are compared in figure 9. The predicted cyclic load is slightly greater than that measured. This is probably due to a slight error in the total mass and mass distribution in the model. The steady component of the chordwise load is the torque producing component. At least part of the difference between measured and predicted chordwise steady load can again be attributed to the difference between the actual rotor speed and that used in the code.



Based on the operation of the steel spar blades on Mod-0, two general observations can be made. First, inasmuch as the measured blade loads (figure 9) were less than those used to predict blade fatigue life, the predicted life is very conservative. Secondly, even though these blades had vastly different mass and frequency characteristics compared to the Mod-0 aluminum blades, they behaved very well on Mod-0. In fact, they appeared to run more smoothly than the aluminum blades even when the tower was in its soft mode.

#### BLADE COSTS

The actual cost of the two steel spar blades is given in table IV. The costs are divided up into three main categories: steel spar and extension piece; wood, foam, fiberglass, etc. material costs; and labor costs for assembly of the airfoil on the spar. These three categories were selected to enable one to project the costs for high rate production.

The actual cost of blades 1 and 2 was \$57,465 and \$49,065, respectively. The major reason for the reduced cost of blade 2 was a one third decrease in the labor required to apply the wood, foam and fiberglass.

The cost of these blades in limited production is estimated to be \$35,000. This estimate is based on a single vendor quote. In high production it is estimated that a blade using this concept could be produced for \$12,500. This includes \$4500 for spar material and labor, \$3000 for airfoil material, and \$5000 for airfoil fabrication.

The above reduction of the spar cost is achieved by assuming the spars are mass produced specifically for wind turbine blades. This eliminates the need for the expensive extension piece and the spar-to-spar weld. Possible means for improving the fatigue life and reducing the weight of the spar are discussed in the next section. The reduction in the wood, foam, etc. cost was achieved primarily by increasing the rib spacing and reducing the Razorback covering from two layers to one layer. Additional cost savings were assumed to accrue from quantity purchases. The labor costs for assembling the airfoil on the spar were reduced substantially based on the assumption that the blades would be mass produced in a highly automated factory.

#### POTENTIAL DESIGN IMPROVEMENTS

The design presented here requires modification because of its limited fatigue life. In addition no attempt was made to optimize the design of the spar. Two potential design improvements are discussed in this section.

##### Double Wall Spar

The fatigue life of steel spar blade could be increased by increasing the thickness of the spar wall where it is welded to the flange. One means of doing this with low cost spar material is illustrated in figure 10. In the highly loaded root area of the blade, the main spar is reinforced by a second concentric spar. This procedure is used routinely by utility pole



manufacturers. The additional thickness of material at the weld should reduce the stress to below the endurance limit. A fatigue test of a double walled spar is planned.

#### Roll Formed Spar

With the roll forming process it is possible to tailor the spar diameter and wall thickness exactly to the load carrying requirements of the spar. A spar formed by this process for a 60-ft. wind turbine blade is shown in figure 11. The spar has a linear taper in diameter and a linear taper in wall thickness except at the weld lands. The spar was designed so that a stress allowable of 30,000 psi not be exceeded for the 120 mph wind case and a stress amplitude of 6000 psi not be exceeded for the 40 mph operating case. It is estimated that a 60-ft. blade using a roll formed spar would weigh less than 3000 lbs.

#### CONCLUDING REMARKS

A 60-ft. wind turbine blade based on a low-cost steel spar as the primary structural member was designed. Two blades were fabricated and operated successfully on the Mod-O wind turbine. Blade loads were close to those predicted, and rotor performance exceeded predictions. Because of the limited fatigue life of the present design, minor design modifications are required to improve the fatigue life. It is estimated that in mass production a blade of this design can be produced for less than \$20,000.

#### REFERENCES

1. Thomas, R. L. and Richards, T. R.: ERDA/NASA 100-Kilowatt Mod-O Wind Turbine Operations and Performance. NASA TM-73825, 1977.
2. Spera, D. A.: Comparison of Computer Codes for Calculating Dynamic Loads in Wind Turbines. NASA TM-73773, 1977.
3. Winemiller, J. R., Sullivan, T. L., Sizemore, R. L., and Yee, S. T.: Design, Fabrication and Initial Test of a Fixture for Reducing the Natural Frequency of the Mod-O Wind Turbine Tower. NASA TM-79200, 1979.
4. Wilson, R. E. and Lissaman, P. B. S.: Applied Aerodynamics of Wind Power Machines. Oregon State University, 1974.

TABLE I. COMPARISON OF ROTOR CHARACTERISTICS  
(NACA 23000 Series Airfoil)

Characteristic	NASA-Steel Spar Blade	Lockheed- Aluminum Blade
Rotor diameter, ft.	126	125
Root cutout, percent	23.0	6.4
Root chord, ft.	6.3	4.5
Tip chord, ft.	2.1	1.5
Root thickness ratio	0.24	0.40
Tip thickness ratio	.24	.12
Solidity	.033	.031
Precone angle, deg.	3.8 (effective)	7.0
Total twist, deg.	0	34 (nonlinear)
Airfoil surface	rib stitched fiber- glass cloth	riveted aluminum
Operating speed, rpm	32	40

TABLE II. MATERIAL PROPERTY DATA

Part		Test no.	Yield stress (0.2%), psi	Ultimate tensile stress, psi	Tensile elong., %
Spar	Flange	1	35,000	59,000 <sup>a</sup>	38 <sup>a</sup>
		2	28,400 <sup>a</sup>	59,600	50
	Weld, flange/pipe	1	35,900	64,600	7 <sup>a</sup>
		2	37,600 <sup>a</sup>	61,600 <sup>a</sup>	7
	Weld, pipe/pipe	1	54,700	79,700 <sup>a</sup>	16 <sup>a</sup>
		2	50,000 <sup>a</sup>	79,800	16
		3	53,900	81,000	16
	Extension piece		---	80,000 <sup>b</sup>	107,000 <sup>b</sup>

<sup>a</sup> Allowable value<sup>b</sup> Estimated from hardness of RC 22

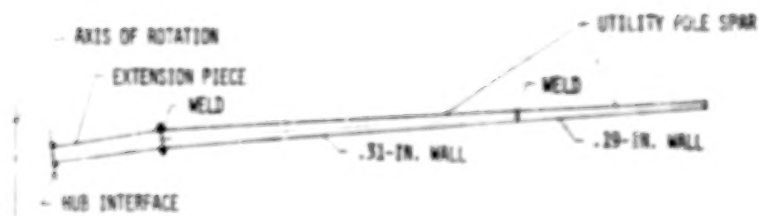
TABLE III. STEEL SPAR BLADE WEIGHT AND BALANCE

Item	Blade No. 1	Blade No. 2
Blade weight, lb.	2460	2488
Balance weight, lb.	68	0
Extension weight, lb.	1089	1125
Total weight, lb.	3617	3613
Spanwise c.g. station, in.	285	285
Chordwise c.g., percent chord	27	27

275

TABLE IV. STEEL SPAR BLADE COST

Item	Blade No. 1	Blade No. 2
Spar Material and Labor		
Main Spar	850	850
Tip Spar	215	215
Flange (forging and machining)	3000	3000
Welding and Inspection	3000	3000
Extension Piece	15000	15000
Total Spar Cost	22065	22065
Airfoil Material		
Wood, Foam, Fiberglass	3000	3000
Razorback and C.A.B. Dope	2000	2000
Total Airfoil Material Cost	5000	5000
Airfoil Fabrication (@\$20/hr.)		
Assemble Wood, Foam, etc.	24000	16000
Apply Razorback, Dope, Paint	6400	6000
Total Airfoil Fabrication Cost	30400	22000
Total	\$ 57465	\$ 49065



(A) STEEL SPAR AND EXTENSION PIECE



(B) BLADE PLANFORM



(C) TYPICAL CROSS SECTION

FIGURE 1 - STEEL SPAR BLADE DESIGN.

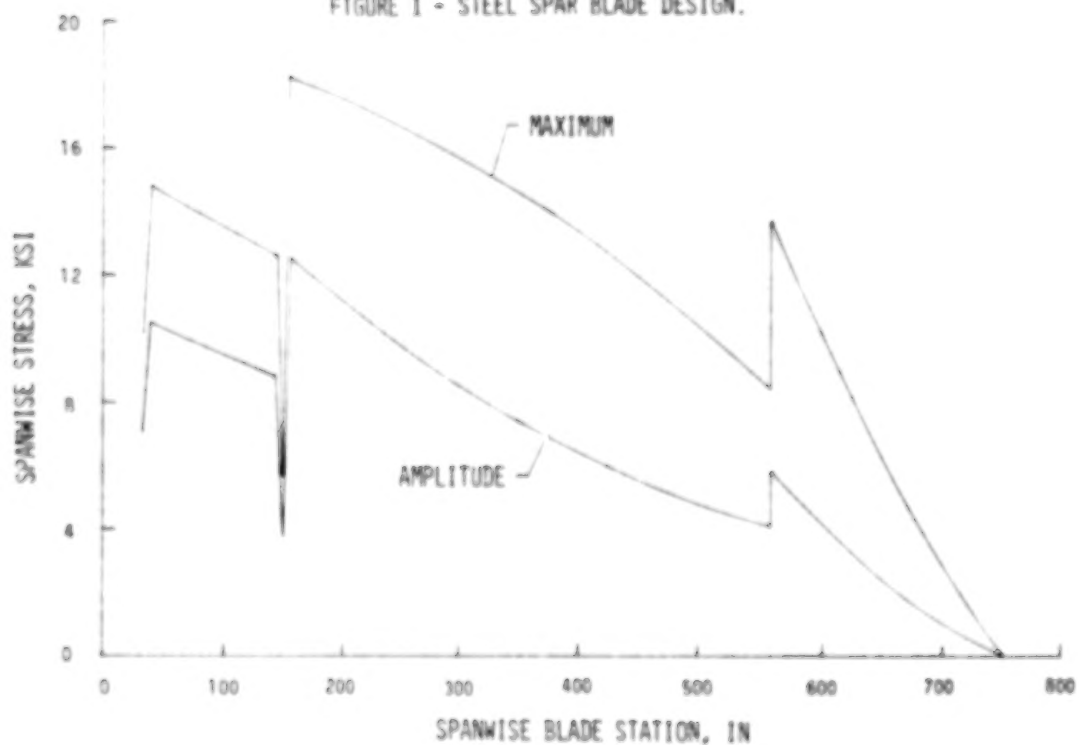


FIGURE 2 - CALCULATED STRESS IN SPAR FOR 40 MPH OPERATION

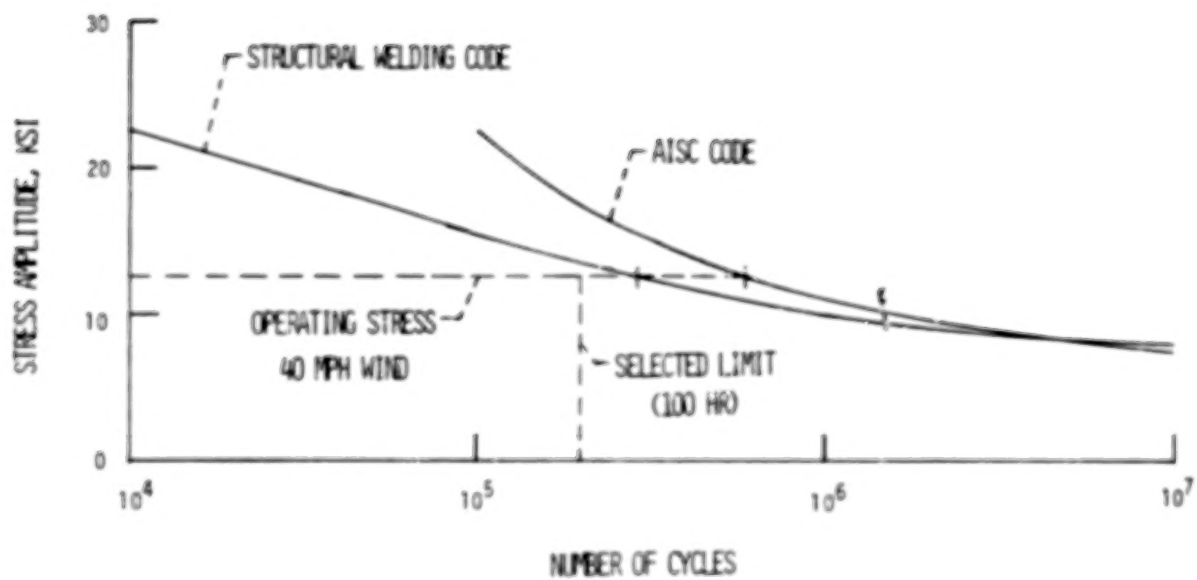


FIGURE 3 - SPAR-TO-FLANGE WELD FATIGUE LIFE ANALYSIS.

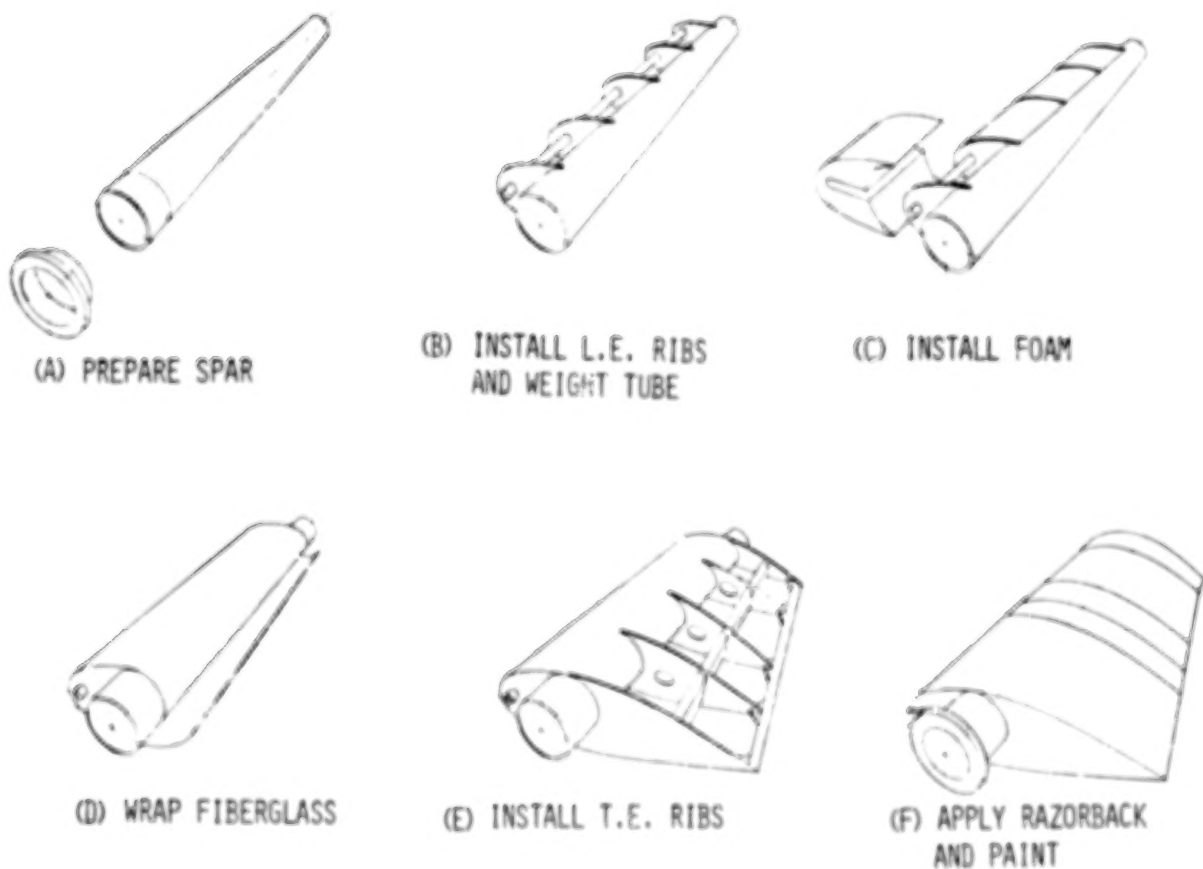
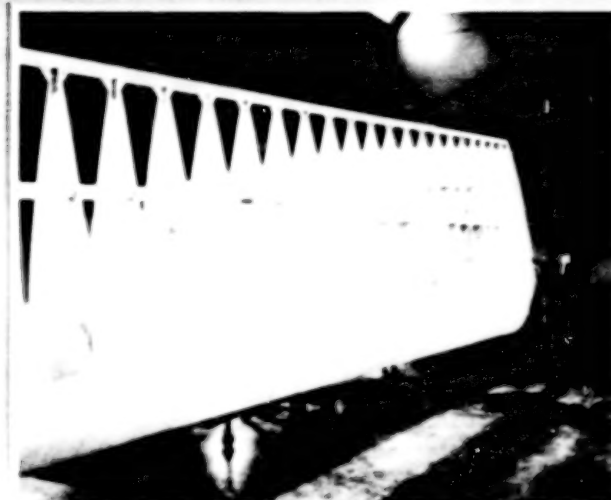
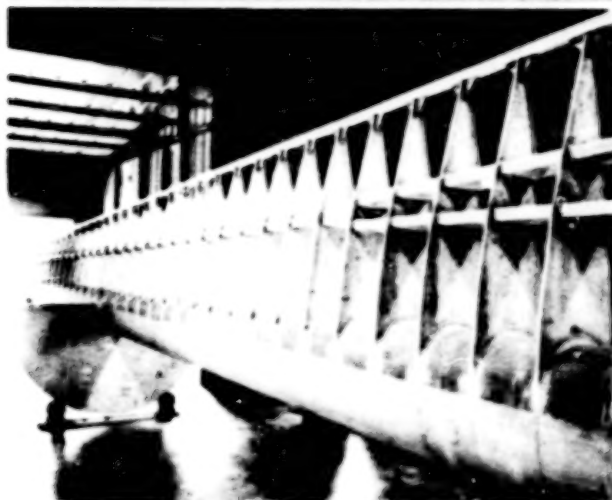
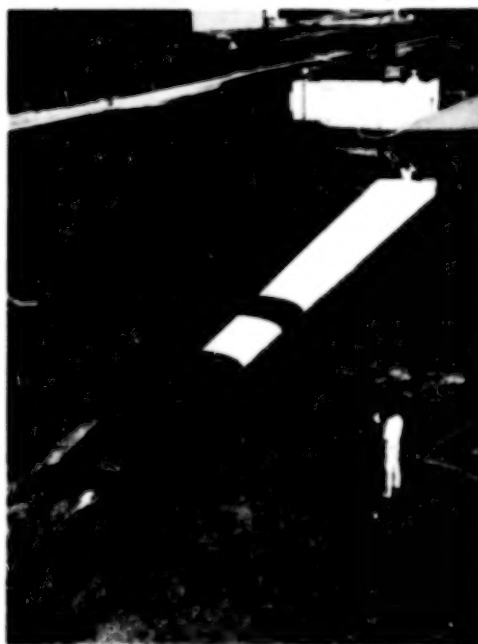


FIGURE 4 - STEEL SPAR BLADE FABRICATION PROCESS.



FABRICATION



LOADS APPLICATION



INSTALLATION ON MOD-O

FIGURE 5 - PHASES OF CONSTRUCTION OF UTILITY POLE BLADES.



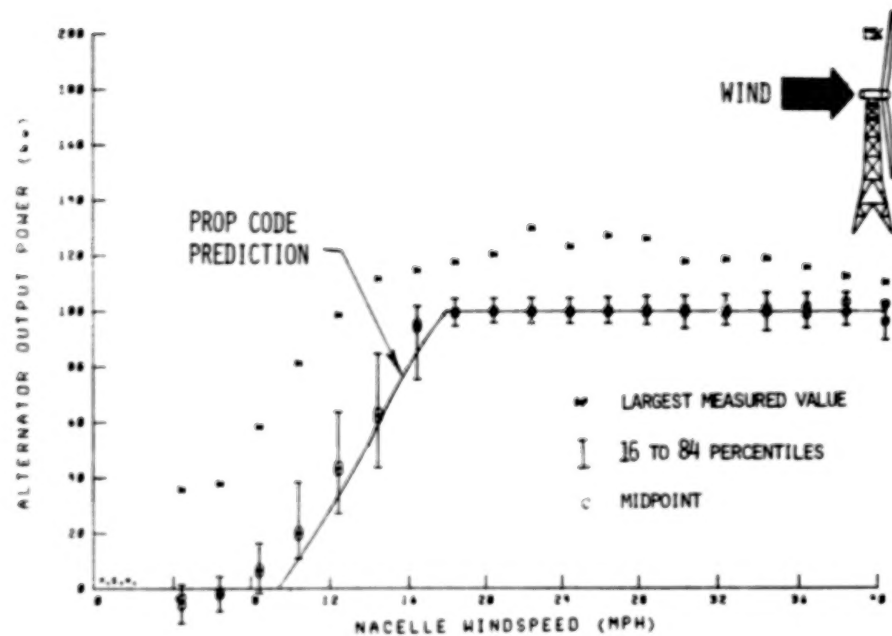


FIGURE 6 - COMPARISON OF PREDICTED AND MEASURED STEEL SPAR BLADE PERFORMANCE (SOFT TOWER).

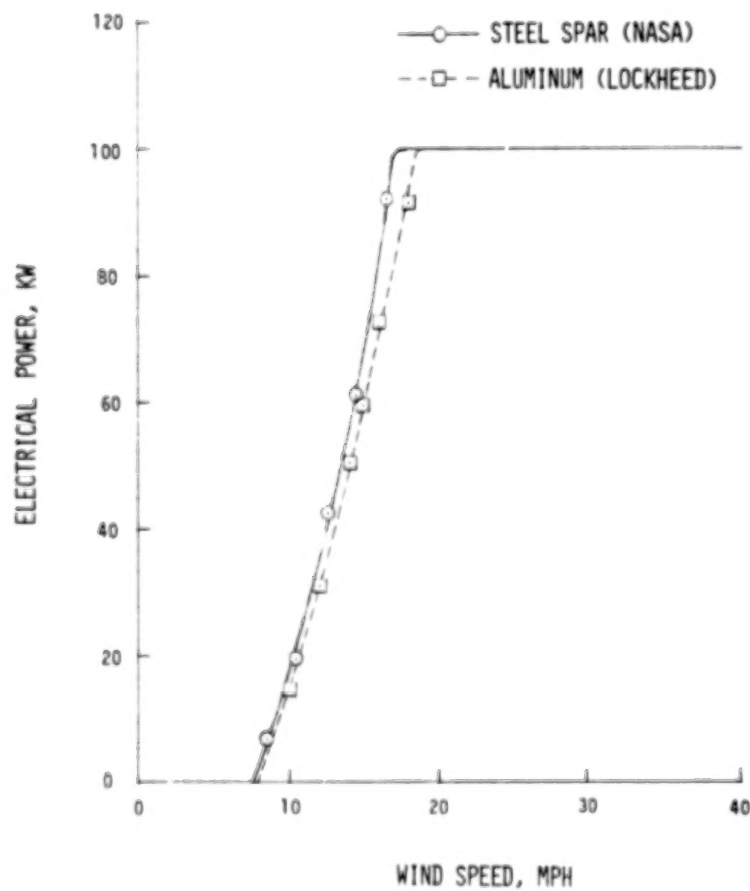
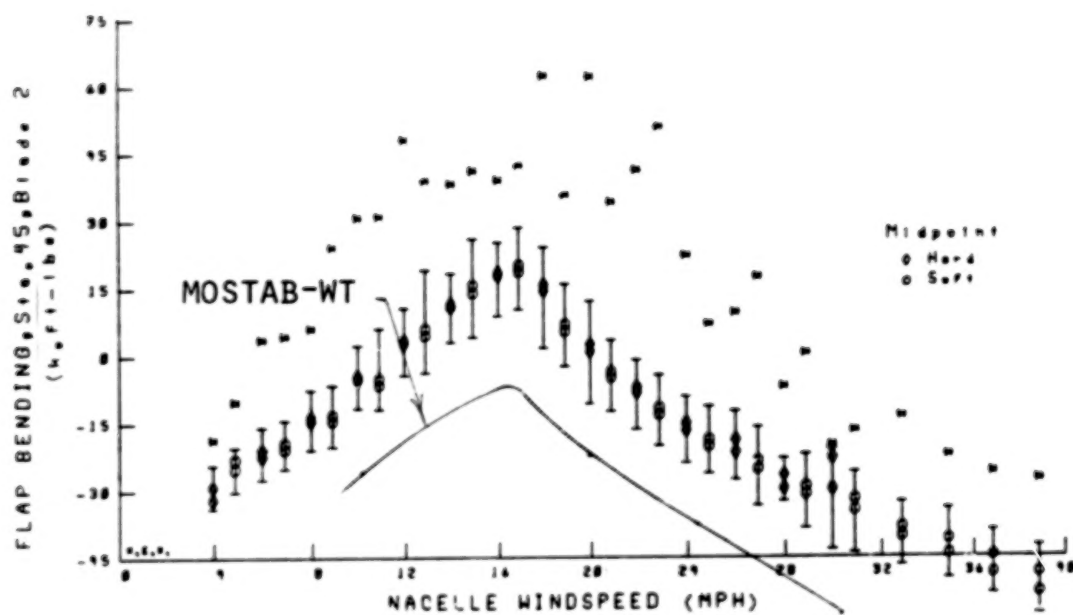
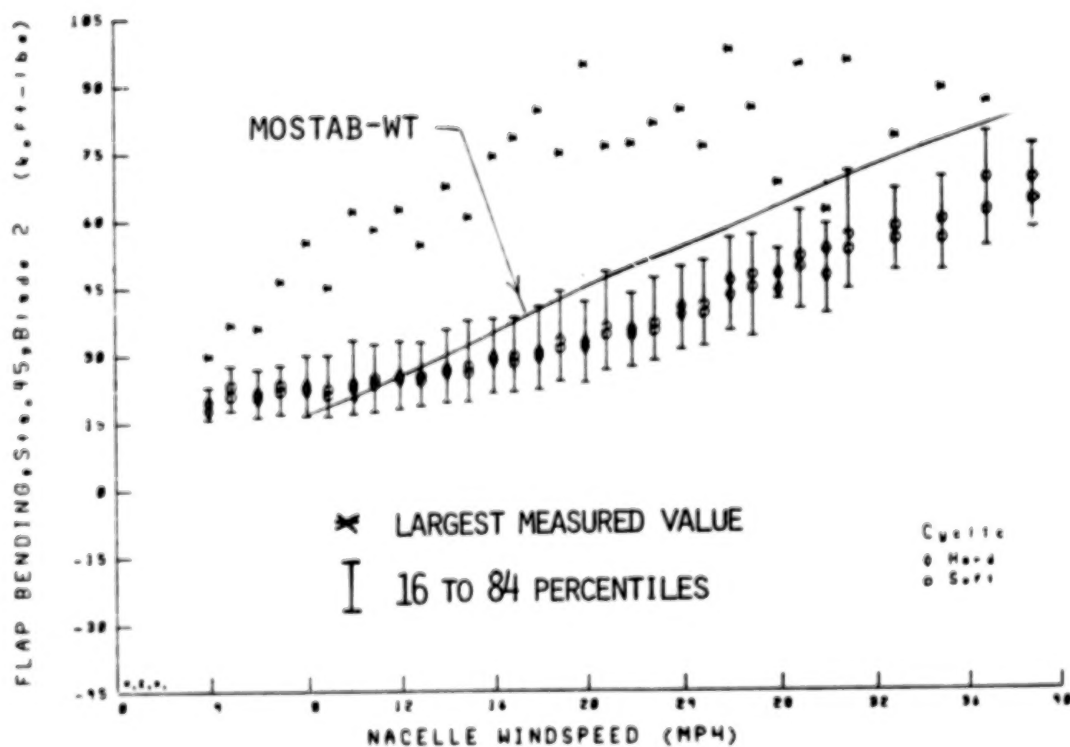


FIGURE 7 - COMPARISON OF PERFORMANCE OF TWO SETS OF MOD-0 BLADES.

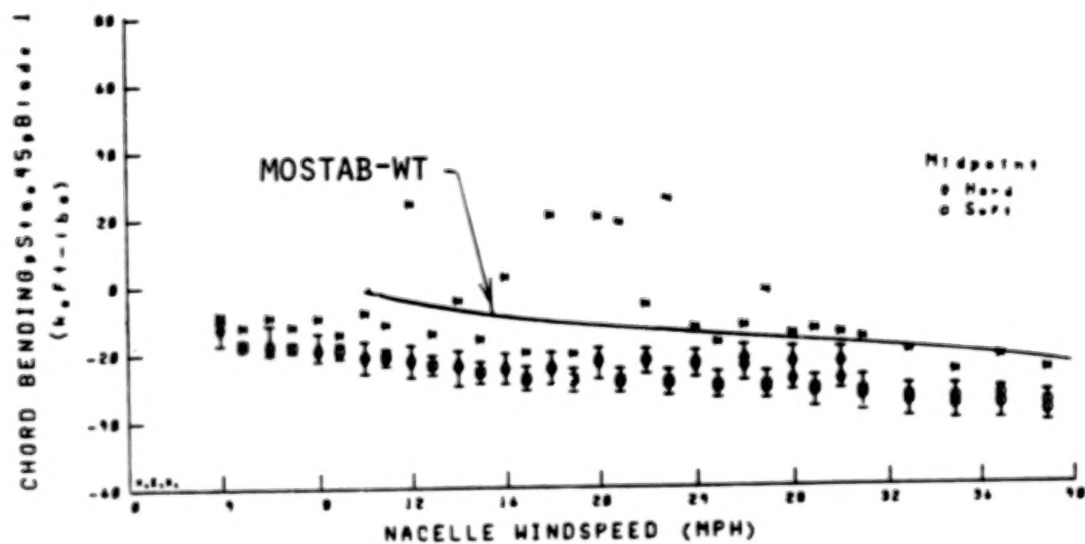


(A) STEADY LOAD

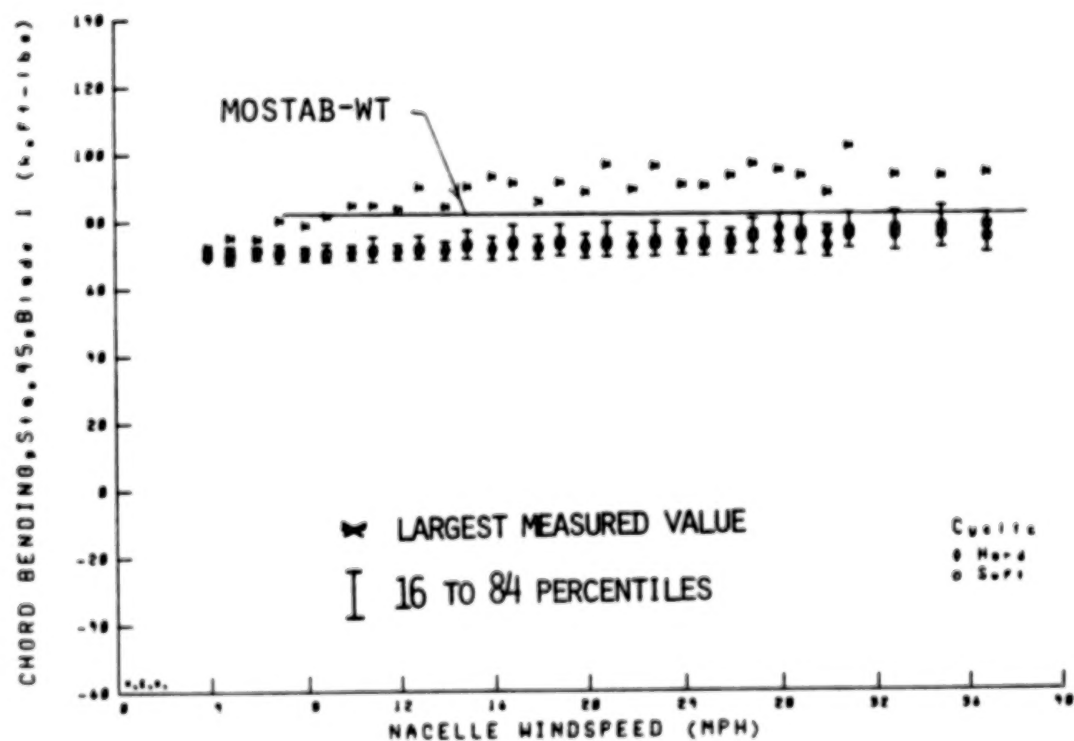


(B) CYCLIC LOAD

FIGURE 8 - COMPARISON OF MEASURED AND PREDICTED STEEL SPAR BLADE FLATWISE ROOT LOADS (HARD AND SOFT TOWERS).



(A) STEADY LOAD



(B) CYCLIC LOAD

FIGURE 9 - COMPARISON OF MEASURED AND PREDICTED STEEL SPAR BLADE EDGEWISE ROOT LOADS (HARD AND SOFT TOWERS).

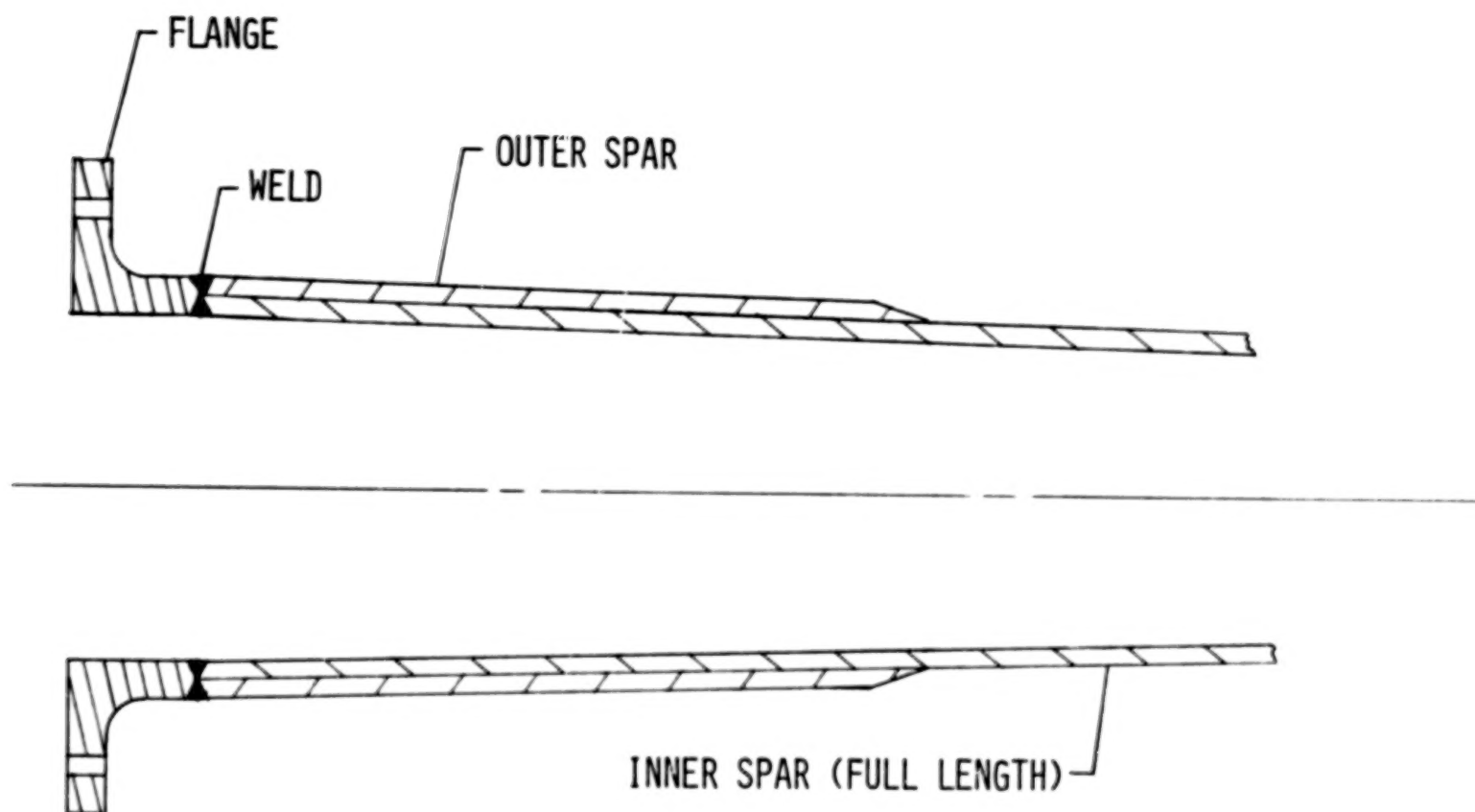


FIGURE 10 - DOUBLE WALL ROOT END SPAR

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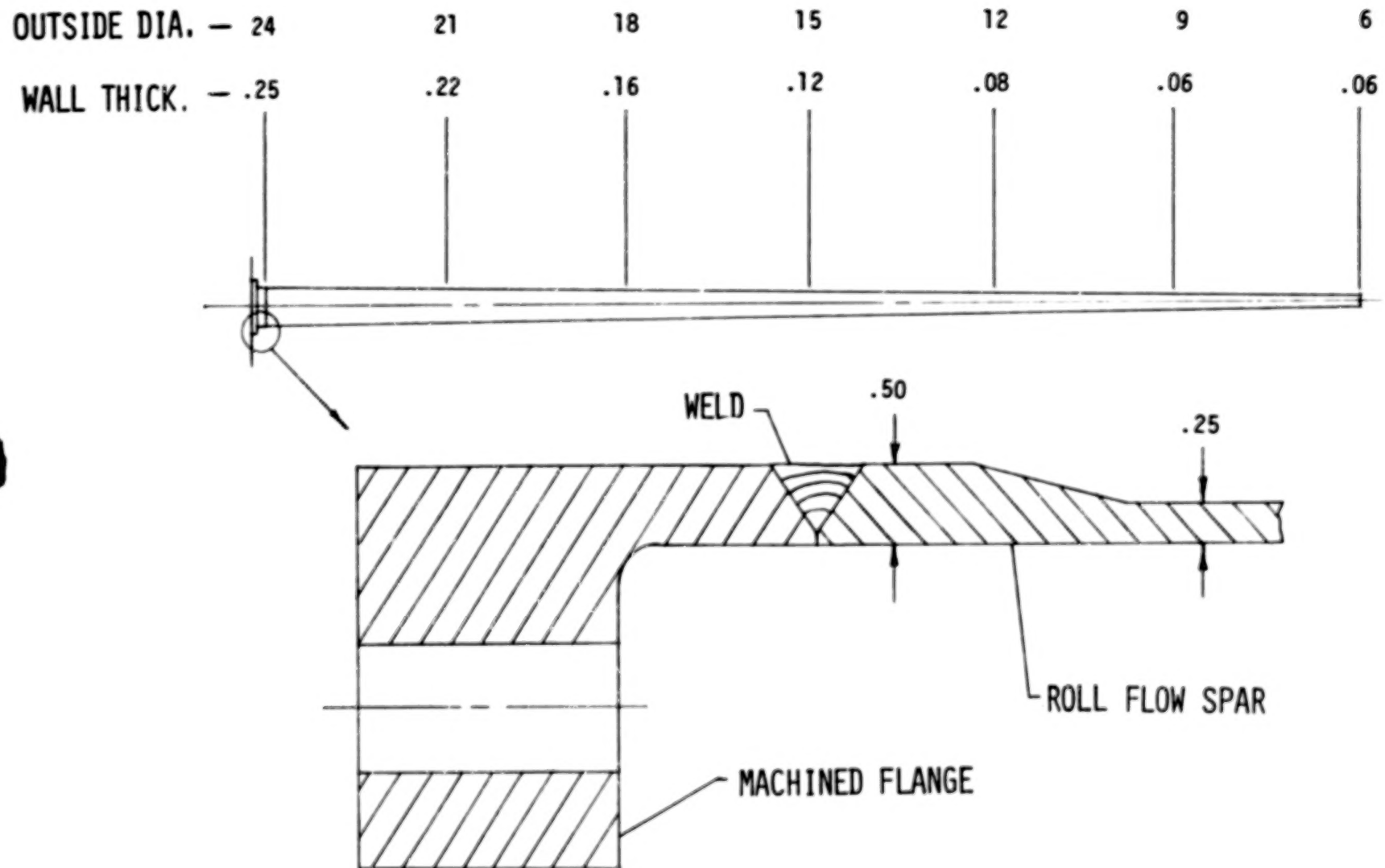


FIGURE 11 - ROLL FLOW SPAR (ALL DIMENSIONS IN INCHES)

## WTG ENERGY SYSTEMS' ROTOR - STEEL AT 80 FEET

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### INTRODUCTION

The 80 foot diameter rotor discussed in this presentation has been installed and operational on WTG Energy Systems' MP1-200 wind turbine generator system since the summer of 1977. The MP1-200 wind turbine is installed as part of the Island of Cuttyhunk's electric power utility grid system. Cuttyhunk Island is located approximately 14 miles off the coast of southern Massachusetts (fig. 1).

The MP1-200 wind turbine was developed, fabricated and installed by WTG Energy Systems as a production prototype. The MP1-200 is a synchronous generating system rated at 200 kilowatts at 28 mph wind velocity. Constant 60Hz, 480VAC current is produced directly from the wind turbine's generator at +/- 1% accuracy throughout the machine's operating range. A micro-processor based control system utilizing electrical load modulation is utilized to maintain constant rotor speed.

The MP1-200 rotor is a fixed pitch design. This configuration was chosen for its low cost and adaptability to sophisticated electronic control systems. As part of the Company's development program, WTG Energy Systems conducted a study of both fixed pitch and variable pitch rotors as speed control devices for synchronous generating systems. This study resulted in a specification for the speed control with a response time of .5 seconds. This criteria eliminated the variable pitch rotor configuration as a rotor speed control device since response time to speed changes caused by changes in wind velocity was far too slow to meet the requirements of synchronous power generating systems.

### MP1-200 ROTOR DESIGN

The MP1-200 rotor shown is of all steel construction for reasons of simplicity, low cost and high inertia. The high inertia obtained by this type of construction greatly reduces the rotor's acceleration due to wind gusting and thus simplifies the control system. The blades are twisted from tip to root 14 degrees. The twist distribution used is the best straight line fit to the analytical exact pitch vs. rotor radius curve. An untwisted blade will suffer badly in the loss of startup capability in a fixed pitch type machine, and running

efficiency in all types of machines. The fixed pitch, constant speed rotor design of the MP1-200 was chosen for its high reliability and cost effectiveness.

The tips of each blade rotate in the pitch plane to an angle of 60 degrees. This function is used primarily as an overspeed shut down device. The tips are held in the run position against heavy duty springs by hydraulic pressure. When the hydraulic pressure is lost the tips are sprung into the shut down position.

The preliminary design of the rotor was predicated on the required power output at 28mph wind velocity, the site's wind regime and energy requirements at the proposed site. The NASA developed GA(W)-1 air foil section was chosen for its high lift coefficient and good start up characteristics. The blade plan form and thickness distribution was chosen so that there is minimum blade interaction. The maximum Reynolds Number was obtained with this configuration. The tip losses due to the rectangular plan form are an important factor except at very high angles of attack. In this operating mode the rotor is already putting out maximum power and the lower efficiency is helpful in keeping the machine from exceeding its name plate rating.

#### ROTOR DESIGN SPECIFICATIONS

The structure of the rotor was designed to withstand many different loads some of which are shown in Figure 2. The major loads involved are the gravitational loads and the hurricane wind loads on the blade spars. The major dynamic vibrational force drivers are the gravity loads and the wind shear loads. The major stresses occur in the rotor spar at the connection of the spar to the hub and the reduction end of the spar section (see fig. 3). In relation to the fatigue sensitive areas the stresses are kept below 15,000 psi. basic stress. In the non-fatigue sensitive areas the stresses are below 30,000 psi. basic stress. The first natural harmonic of the blade perpendicular to and in the plane of rotation is over 2.5 cps.

The blade spars are constructed of SAE 1026 seamless steel tubing, the stamped reinforced ribs are of 18 Ga. steel, welded to the spar on 12 inch centers, and the blade skin is 18 Ga. galvanized sheet steel riveted and bonded to the ribs. The hub is cast in two sections. The hub material is of high toughness, 60-40-18 ductile iron. The pitch of the rotor is set to produce maximum efficiency at approximately 2 mph higher than the installation site's annual mean wind velocity (16.9mph @ Cuttyhunk Island). The rotor blade pitch is set at the time of field assembly. The pitch is maintained by the clamp fit of the blade spars installed between the two halves of the hub.

## ROTOR PERFORMANCE

Within the operational envelope it can be shown that the fixed pitch rotor on the MP1-200 has a power coefficient of over .4. The reason behind this is at low wind velocities the blade angle of attack to the relative wind is minimum, thus a low drag factor exists. In high winds the blade is working at a high coefficient of lift, high angle of attack and has the torque to overcome the higher drag related to this operating condition.

The fixed pitch design also allows us to use the stalling effect above the system's rated wind velocity to limit the power output. Blade stall has proved not to affect a rapid change in power output, but, rather a gradual loss of efficiency, as desired. This affect can be seen in Figure

The three blades have been used in order to improve the startup characteristics of a fixed pitch rotor. As shown in Figure 4, start up and power production regularly occur in an 8 mph wind.

## COST

The sale price FOB the plant in Buffalo, New York as described, is approximately \$40,000. This price represents approximately 18% of the total cost of the MP1-200 system. The cost breakdown of the rotor, by components, see Figure 5, shows that the two most expensive parts are the rib and hub fabrication, each accounting for 18.4% of the total rotor cost.

The largest single cost is for labor, accounting for 26.8% of the total. The labor rate is \$18.90/hr (1978 \$). In an analysis of the production and design of this rotor system, WTG Energy Systems concluded that, with a limited production schedule of five units tooling, material and fabrication costs could be cut by 20%, and assembly labor costs cut by 30%. Therefore the potential savings for a production run of five units would be 50% for a total FOB price of approximately \$20,000 (all dollars are assumed to be 1978 \$), or \$100 per rated kilowatt.

## FUTURE R & D REQUIREMENTS AND SUGGESTIONS

The areas in the design and operation of the MP1-200 rotor system, in particular, and fixed pitch rotor systems, in general, requiring additional research and development are listed below.

1. A detailed analysis of the stalling characteristics of large, fixed pitch rotors should be carried out. This should include field testing to document the rotor thrust and torque forces, and, rotor vibrational dynamics from startup,



pre-stall, to the fully stalled rotor and beyond to the machine's rated cut-out wind velocity.

2. During the field testing of the MP1-200 system it has become apparent that the rotor operation has some stabilizing effect with regard to the dynamics of the tower. A detailed analysis of the gyroscopic effect(s) of large rotors on tower stabilization should be carried out. This could lead to significant reduction in the cost of both the tower and foundation for large wind turbines.

### CONCLUSION

To date the rotor has seen well over one million, fully reversing, maximum load, fatigue cycles in the maximum stress areas with no component failure or signs of metal fatigue. Since the stresses caused by the in plane gravitational loads are by far the most significant, we feel confident that the MP1-200 rotor system, as installed on Cuttyhunk Island, is a fatigue resistant design. The rotor on the MP1-200 has been subjected to wind velocities in excess of 100 mph on four separate occasions. Neither the rotor nor any part of the wind turbine has suffered any damage as a result.

The rotor has been subjected to overspeed conditions of approximately 70% of its rated speed (80 rpm) resulting in no adverse effects on the rotor, drive line components or support systems.

The rotor has demonstrated good startup characteristics and power production in 8mph wind velocities.



Figure 1. - MP1 - 200 wind turbine generator. (Photo courtesy of Eagle Signal Division, Gulf and Western Manufacturing Company.)

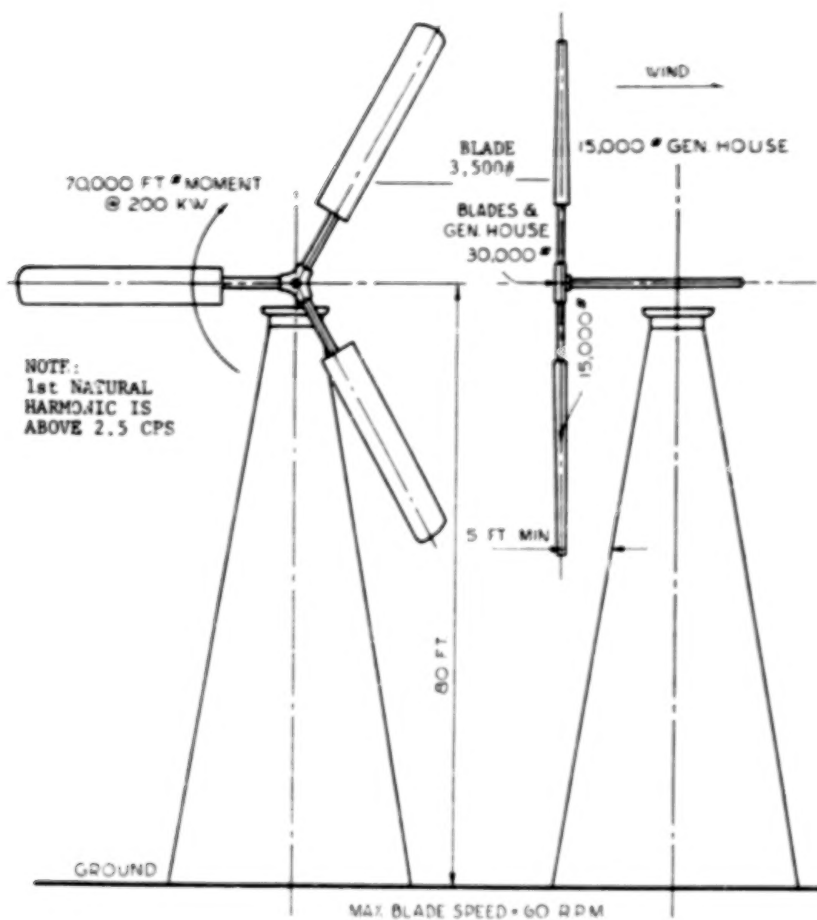


Figure 2. - Loads.

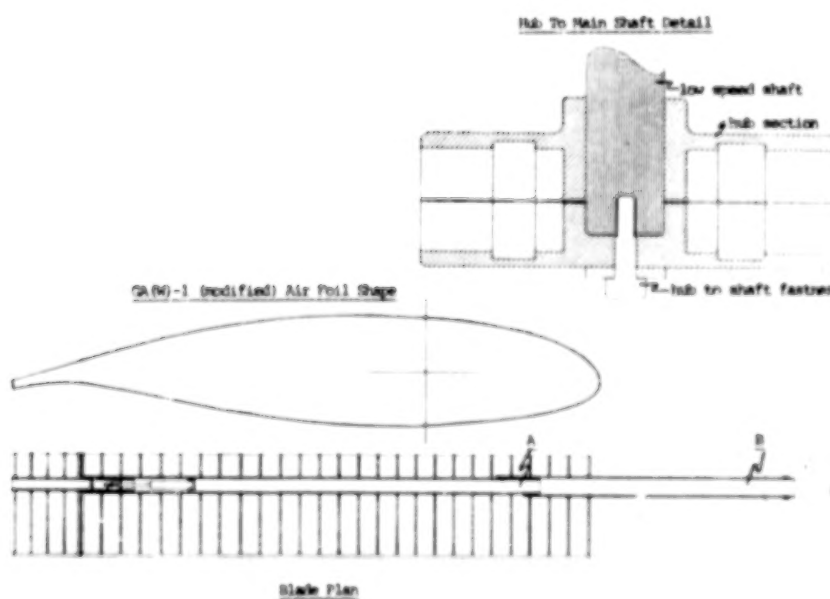


Figure 3. - Rotor assembly plan.

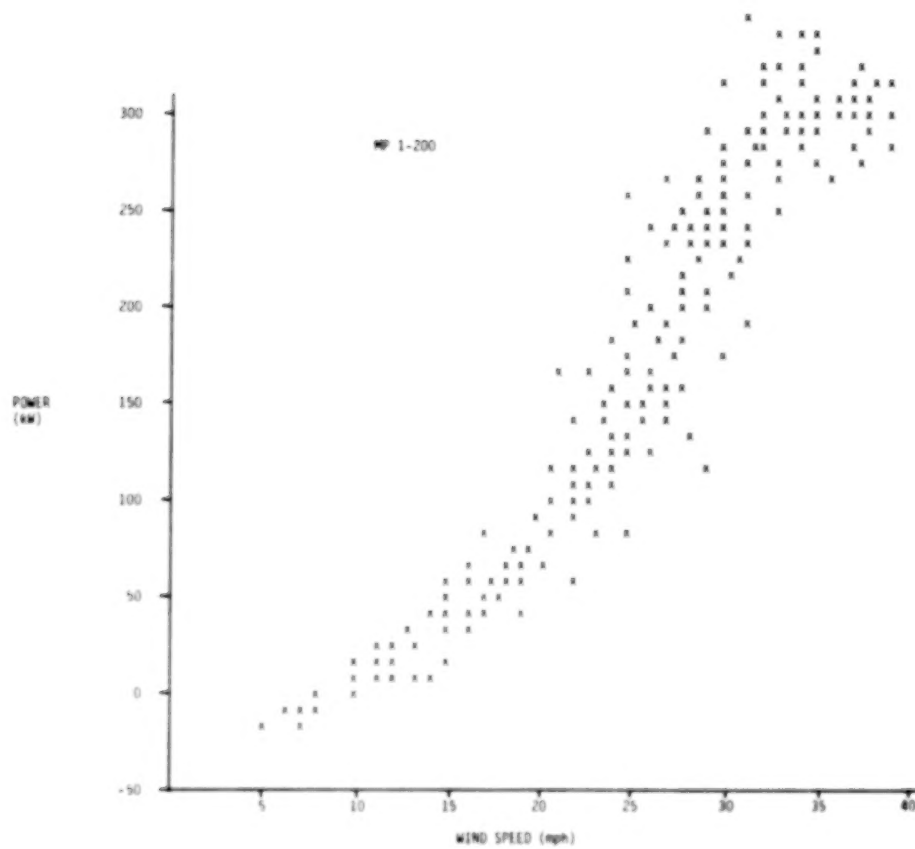


Figure 4. - Generator output curve.

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<u>COMPONENT</u>	<u>PER CENT OF TOTAL COST</u>	<u>COST</u>
RIBS	18.4%	\$ 7,560.00
SPARS	12.3%	5,040.00
HUB	18.4%	7,560.00
TIP FLAPS	13.8%	5,670.00
BLADE SKIN	5.4%	2,250.00
MISC. HARDWARE	4.9%	2,030.00
LABOR	26.8%	10,810.00
<u>TOTAL</u>	<u>100%</u>	<u>\$ 40,920.00</u>

Figure 5. - Cost breakdown MP-1 200 rotor system by major components (sale price for limited production).

## THE USE OF WOOD FOR WIND TURBINE BLADE CONSTRUCTION

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### Historical Problems With Wood and a Modern Solution

With the development of modern engineering materials, such as steel, aluminum and fiber reinforced plastic composites, the use of wood as a serious engineering material for sophisticated structures has greatly diminished. The reasons for this are generally well known; wood can deteriorate from rot, and be dimensionally unstable. Then the fact that the consistency of wood can vary within a single tree together with fluctuation in physical properties because of moisture level change provides for difficult quality control problems.

We feel that the demise of wood as a serious engineering material is both unfortunate and premature. With the help of modern technology, most of the problems with wood can be solved in a practical manner. For the past 10 years, we have used wood in composite with plastic resins to build high-performance sailing craft; specifically, iceboats and multihull craft that must be built at high strength-to-weight ratios to be successful. In part, our success has been due to the fact that wood itself is an excellent engineering material, and in some applications has capabilities that are unavailable with any other material (which we will explain later on in this paper). Our ability to solve the moisture problem with wood, however, was the key to the development of wood as a practical engineering material even for use in a hostile marine environment.

To better understand what we have done to achieve our solution, a discussion of the interrelationships between moisture and wood is needed.

Moisture is a major ingredient of all wood in the living tree. Even wood that is properly dried or cured will have a significant percentage of its content by weight being moisture. This will typically range from 8% to 15% of the oven dry weight of the wood, depending upon the atmospheric conditions in which the wood exists. Figure 1 shows the ultimate moisture content of wood when subjected to various relative humidities at a temperature of 70°F. Unfortunately, the subject is a little more complicated than the chart portrays because 50% relative humidity is much different at 40°F. than at 70°F. (warm air holds more moisture than cold air), but every area will have an average year around moisture and temperature level that will determine the average wood moisture content level. In our Great Lakes area, wood seems to equalize at about a 10% to 12% moisture content when dried in a sheltered but unheated area. The real problem with wood, is that its moisture level is rather quickly influenced by short term changes in atmospheric conditions. In the Great Lakes area, we continually have extremes of dry and humid climate conditions that are compounded by temperature extremes of 100°F. between winter and summer.

Wood cells are quite resistant to the invasion of moisture in a liquid form, but moisture vapor as a gas has a sudden and dramatic effect on wood by being able to easily and quickly pass through the cell wall structure. Responding to the changes in atmospheric conditions, unprotected wood may undergo many moisture changes in a short period of time, and the repeated dimensional expansion and contraction of the wood under these conditions is thought to be the leading cause of wood to age prematurely.<sup>1</sup> Conversely, wood in its natural state as a living organism will remain at a relatively constant moisture level during its entire lifetime until it is harvested.

This sponge-like capacity to take on and give off moisture at the whim of the surrounding environment in which it exists, is the root cause of all of the problems with wood. Specifically, varying moisture levels in wood are responsible for: (1) dimensional instability; (2) internal stressing that can lead to checking and cracking of the wood; (3) potential loss of strength and stiffness of the wood; (4) wood decay due to dry rot activity.

Dimensional instability has always been a limiting factor for the use of wood in many engineering applications where reasonable tolerances must be maintained. To complicate matters, the dimensional instability of wood has never been constant and varies widely between species of wood, with radial grain wood (cut perpendicular to grain) in most species being more stable than is tangential wood (cut parallel to grain). The dimensional change of wood due to moisture change always occurs first on the outer surface causing differing moisture levels to occur within the same piece of wood. This can lead to internal stressing that often is the cause of checking and cracking on the wood surface.

Moisture has a significant effect on the strength of wood. Dry wood is much stronger and stiffer than is wet wood. The reason for this is the actual strengthening and stiffening of the wood cell walls as they dry out. If wood is taken at its fiber saturation point of 20% and allowed to dry to 5% moisture, its end crushing strength and bending strength may easily be doubled and in some woods tripled. The result, is that wood has the potential to be an excellent engineering material when dry, but only a mediocre material when wet. This causes a vexing problem for the engineer who may not be able to determine the level of moisture content that can be maintained in the structure he is designing, and must assume a worst case situation.

Of all of the problems with wood, dry rot decay is the most known and feared. Dry rot is a misleading term since dry wood does not rot, there

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<sup>1</sup> Wood has been taken out of the tombs of Egypt that has been over 3,000 years old. Because of the constant temperature and humidity in which it was stored, the wood has lost none of its original physical properties.

in fact must be rather exact conditions in order for dry rot spore activity to occur: (1) The moisture content of wood must be at or near the fiber saturation point of 20%. (Rot is unknown in wood with a moisture content less than 20%)., (2) There must be an adequate supply of oxygen available to the rot spore fungi, i.e. the wood must not get too wet, (3) The temperature must be warm, 76° to 86°F. is ideal, although fungi have been known to be active at temperatures as low as 50°F. (4) There must be the proper kind of food. Some woods such as western red cedar are resistant to rot because of the tannic acid in their cellular makeup.

Although there are many types of rot fungi that can destroy wood, in North America there are two species of the brown rot family that predominate. They are very hardy creatures that seem to survive the worst large temperature extremes in a dormant state waiting only for the right conditions to occur to become active. Efforts to control the brown rot family have had only limited success and generally center around poisoning the food supply with various commercial wood preservatives. Our approach to solve this problem is quite different as will be explained.

#### The Wood Resin Composite

As we have discussed, most of the problems that we have with wood are moisture related. Therefore, a primary goal of incorporating wood into a composite with a resin is to provide maximum protection against moisture to the wood fiber. Our basic approach is to seal all wood surfaces with our proprietary resin system. This same resin system is used as the bonding adhesive to make all joints and laminates with the goal that they too will be secured from moisture. The lamination itself, can usually be counted upon to offer further moisture protection. To build our structures, we usually laminate thin veneers together and the glue line between each veneer serves as a significant moisture barrier. For instance, when using 1/16 inch thick veneers in a 1 inch thick laminate would provide 15 glue lines that must each be penetrated to increase or decrease the moisture content of the entire laminate.

The success of the wood resin composite method depends on the ability of the resin to resist moisture passage. Our resin system is the most effective moisture barrier that we know of and has proven itself through actual useage over many years in marine construction. We cannot claim that our resin system forms a perfect moisture barrier, but it does slow the passage of moisture into the wood to such a great extent that any moisture change within the wood itself is minimal. If dry wood encapsulated in our resin were put in a steam bath and left there for several months, the moisture in the wood would eventually rise. However, the rate of moisture change in a wood-resin composite is so slow under normal changing atmospheric conditions that the wood inside remains at a virtually constant moisture level that is in exact equilibrium with the average annual humidity and is able to easily resist violent seasonal moisture fluctuations. With the moisture content of the wood stabilized at lower levels, we are able to maintain good physical properties together with excellent dimensional stability. Dry rot is eliminated as a problem by keeping the



moisture content below that required for dry rot activity and also by completely sealing the wood from an oxygen source that is a necessary ingredient for the rot spores to survive. Our testing has shown that even if wood should reach a moisture level high enough to support rot spore activity, the rot spores still cannot exist without adequate oxygen.

#### Structural and Economic Considerations

We, of course, did not invent the principle of laminating wood; this process had been commonly used for a number of years. There are, however, some significant differences with our method. First, a wood-resin composite laminate as we would build it is composed of a very high resin content by weight, considerably higher than what is considered normal in the general wood laminating industry. This high percentage of resin-to-wood ratio is desirable for several reasons. Enough high-density plastic is available in the composite to provide sufficient moisture protection to all of the wood fiber. Our resin also has excellent physical properties with the potential to improve the composite structurally. Wood is considerably stronger in tension than it is in compression. The resin that we have developed is just the opposite, being much stronger in compression than it is in tension. By properly mating the two materials, one complements the weakness of the other with the potential for more strength than either would be capable of on its own.

Wood laminating is usually accomplished at pressures of 75 to 100 psi to make effective bonds. Achieving these high pressures, can be very expensive especially in overhead and capital expense for tooling. High pressures also severely limit the size of the laminated part that can be made. With our resin, we are able to make perfect bonds at low pressures. In many cases, only contact pressure is needed between wood pieces because our resin has sufficient physical properties to easily span small gaps if they should occur. Lowering the pressure needed for laminating has the positive effect of lowering the cost of wood bonding. Pressures of up to 12 psi are easily and cheaply developed with the use of the vacuum bag system and are sufficient to manufacture all of the laminated parts for the 60' wood composite wind turbine blade. Thus, the cost of molds and mixtures are quite inexpensive allowing both low or high unit production to occur at low per unit costs.

#### Quality Control

Using a high-strength adhesive for bonding reduces significantly quality control problems from those normally associated with the wood laminating industry. The physical properties of our resin are considerably higher than the grain strength or the shear properties of the wood. This excess structural capacity of the resin adhesive provides a wide safety margin that has proved extremely important to our success in the manufacture of lightweight boats. We have been able to produce on a regular basis, highly reliable bonded joints between wood members with only contact pressure. Even significant voids that might develop between wood laminates

do not present a problem provided that there is sufficient resin adhesive to span the void.

Using a single piece of wood in a critical application has always posed a difficult quality control problem. An experienced individual had to carefully inspect each piece used for hidden flaws that might compromise strength. The multipiece lamination solved this problem by using the "safety in numbers" principle. In our turbine blade application, there will be up to 40 laminations of 1/16 inch thick veneer to form the main load-bearing "D" section. Even if several of these laminations were flawed and slipped by inspection, it would have little effect on structural capability of the entire lamination.

#### Wood as an Engineering Material

In considering wood as an engineering material, it is pertinent to note that "wood" is not a single material with one fixed set of mechanical properties, but rather includes many species which possess a wide range of properties, which depend upon both the species and the density selected. The range of properties is considerably wider than what is generally available with most other types of materials, where some variation of properties can be attained by means such as alloying or tempering, but where little variation of material density is possible. Wood, on the other hand, can be selected over somewhat more than a full order of magnitude in density, from 6 lbs/cuft or even less for selected grades of balsa, to over 60 lbs/cuft for certain species of hardwood. The flexibility this can provide the wood structure designer is obvious; since low-density species can be selected for efficient use as core materials, or for panels or beams where stiffness or buckling resistance is of primary importance. High-density species can be selected where there is a need for high strength and modulus per unit volume, such as in panel skins or in structural members which must occupy constrained geometric volumes. The full range of intermediate densities provide a match for requirements anywhere between these extremes. In this regard, it is worth noting that the physical properties of wood are roughly proportional to its density, regardless of species, since the basic organic material is the same in all species, and thus changing density is rather like compressing or expanding the net strength and elastic stiffness into different cross-sectional areas, with little net variation of total properties per unit mass (table I).

Given that the strength and modulus of wood vary approximately in proportion to its density, it is easily shown that the length of a solid wooden panel which is stable against buckling will vary inversely with its density, while the length of a solid wooden column stable against buckling will vary inversely with the square root of its density. Therefore, approximately a factor of ten in unsupported panel length, or a factor of

three in unsupported column length, is readily available to the designer of wooden structures. Designers of structures using other materials can perhaps best appreciate what this means by imagining that a factor of 10 of density variation were somehow readily available for the steel, or aluminum, or composite, with which they regularly work.

Granted that the density variation of wood can be of advantage to the wooden structure designer, one must also inquire how good are its net properties per unit mass relative to other structural materials. There are, after all, other light variable density materials available, such as the expanded foams. For modern structures where weight is an important issue, the strength-to-density ratio (specific strength) and modulus-to-density ratio (specific modulus) are two very important numbers to consider, since they determine how much strength and stiffness you can get for a given mass of material.

A typical grade of Douglas fir, a moderate density species, will possess approximately the following properties:

#### Fir

Density	.52 (32.5 lbs/cuft)
Compressive Strength	7500 psi
Tensile Strength	15,000 psi
Modulus	$2 \times 10^6$ psi

To easily compare this to other materials, the table below indicates the strength and modulus required of the other materials to achieve exactly the same strength-to-weight, and modulus-to-weight, possessed by Douglas fir.

#### Equivalents of Fir

	<u>Steel</u>	<u>Aluminum</u>	<u>Fiberglass Composite</u>
Density	7.8(487 lbs/cuft)	2.7(169 lbs/cuft)	1.9 (119 lbs/cuft)
Compressive	112,500 psi	38,942 psi	27,403 psi
Tensile	225,000 psi	77,885 psi	54,807 psi
Modulus	$30 \times 10^6$ psi	$10.38 \times 10^6$ psi	$7.3 \times 10^6$ psi

Those familiar with the typical properties of steel, aluminum, or fiberglass composite, will recognize that these numbers indicate Douglas fir to be a competitive structural material on a per unit weight basis. It might also be noted that the number cited for the fir do not represent unusually good samples or unusually dry samples, and typical shop laminates we have produced, in fact, exceed the strength and modulus numbers cited.

It should be pointed out at this time that the preceding considered the properties of wood along its grain direction. The same piece of fir which displays 15,000 psi tensile strength along its grain direction will have something like 300 psi maximum tensile strength across its grain. That is

a 50 to 1 variation in tensile strength depending on the load direction. The other physical properties of wood are also distinctly anisotropic, although not to as great a degree. What this means is that the wooden structure designer may have to take explicit measures to deal with cross-grain or shearing forces within the wooden structure which could safely be regarded as negligible by the designer who uses a conventional material with isotropic properties, such as steel or aluminum. It also means that in cases where large loads flow in more than one direction, that wood fiber will have to be arranged to align with all of these loads. For cases where the large loads are confined to a single plane, a structure such as laminated veneer or plywood can meet the requirements. Where loads in all three axis exist, the designer must use more sophisticated approaches tailored to the loads and geometry. All these factors are the other side of the wooden structures coin, and dealing with them is the price which the wooden structure designer pays in order to gain the advantages of this easily fabricated, high-performance, low-density structural material.

A final factor which must be considered when evaluating wood as a structural material concerns its performance in fatigue. By its very nature as a fibrous material, wood is not given to the kind of fatigue crack propagation that is familiar with metals. The literature of the fatigue properties of wood is not as well developed as that of other materials, but in round numbers, one can expect essentially infinite fatigue life for wood with loads at 30% of ultimate. For some kinds of loading, even higher percentages are acceptable. When one considers that nature has spent millions of years in the serious business of competitive survival to develop good strong trees, which must stand repeated and highly variable loads from winds and other load sources, it should not be too surprising to find that wood is an efficient structural material with very respectable fatigue properties. In fact, one should note that a tree is basically a cantilever type structure subjected primarily to variable wind loads, and that it grows in such a way that the major forces do flow along the grain direction within the tree. Since it happens that modern wind turbine blades are also cantilever type structures subjected to variable wind loads, it is not surprising that wood should be considered potentially advantageous for such an application.

#### Wood Wind Turbine Blade Feasibility

In order to investigate the feasibility of a wooden wind turbine blade, NASA/DOE awarded a small contract to the Gougeon Brothers, Inc., in November of 1977. Several construction concepts were considered and evaluated. A monocoque "D" section forming the leading edge, and a built-up trailing edge section was the selected method of construction. The required thickness to achieve the necessary structural properties in the "D" was examined for both a laminated veneer and bonded sawn stock fabrication technique. Both of these techniques were ultimately judged to be feasible, with the comparative fabrication advantages determined by blade size and special epoxy and wood stock handling techniques, rather than by the resultant physical properties of the finished nose.

In attempting to achieve a practical tail construction with a center of gravity for the blade at the quarter chord point, a number of tail panel construction techniques were considered. These included: (1) simple ply supported by stringers; (2) fiberglass/foam/fiberglass; (3) plywood/honeycomb/plywood; (4) plywood/honeycomb/plywood with aft web and slotting to relieve tail buckling. The final results of detailed strength and stiffness calculations for the last tail panel configuration showed that it was indeed feasible to use wood to meet the Mod OA blade structural requirements. This work is presented in detail in the final report for NASA contract No. DEN3-9. A summary of the basic blade parameters is given in figures 2 through 7.

The projected blade weight and center of gravity location which resulted from this feasibility study were quite encouraging, indicating that a blade under 2,000 lbs, with a center of gravity location reasonably near 25% chord, could be produced using wooden construction techniques.

As part of the wooden blade design, a somewhat unusual but very simple method was proposed to attach the root end of the blade to the hub. The proposal was simply to epoxy bond 24 steel studs into the 3-inch thick wood buildup which exists at the root end (see Figure 8). While the potential economic advantage of such a simple construction was clear, its engineering viability was perhaps not so clear, (even though similar techniques had been successful in a wide range of sailing craft already built and tested), and therefore test samples of these wood to steel stud bonded joints were fabricated by Gougeon Brothers and tested by NASA-Lewis. The results of these tests are available in our final report under contract No. DEN3-9 and show the engineering viability of the direct bonding technique both for withstanding maximum onetime loads, and also for withstanding repeated cyclic fatigue loads. A onetime load in excess of 40 tons was achieved for one of the samples using a 15-inch long, 1-inch diameter stud, to give you a feel for the load transfer which is possible. In the fatigue testing, the 1-inch diameter steel stud often failed before the wood or epoxy bond. This stud bonding is considered to be a very good example of the simplifications which are possible using advanced wood/epoxy construction techniques.

As a final test of wood/epoxy construction for MOD OA wind turbine blades, a 20-foot root end sample was built to the dimensions indicated in the feasibility study, complete with 24 bonded-in studs. This sample has been successfully subjected to large onetime loads in both the flatwise and edgewise directions in tests performed at NASA Lewis. It has also been subjected to fatigue testing at the Fort Eustis Applied Technology Laboratory USARTL (AVARCOM) test facility in Virginia.

Photographs of various phases of the construction of the test blade sample are shown in figure 9.



The results to date indicate that wood is both a viable and advantageous material for use in wind turbine blades. Its low density simplifies the provision of adequate buckling strength for the walls of the blade structure. Both its natural fibrous composition and its ability to be readily bonded into a virtually monolithic structure contribute to the prospect of excellent fatigue life. The quite good physical properties on a per unit weight basis allow a reasonably light blade which is still strong and stiff enough to meet operational requirements. In addition, the basic material is reasonably priced, domestically available, ecologically sound and, most important, easily fabricated.

#### The Cost of Wood

Douglas fir is the chosen material to manufacture the "D" section which makes up approximately 70% of the blade weight. This species of wood was initially considered due to its excellent physical properties, but became the favored material because of availability and low price. With modern reforestation programs, the Douglas fir species is being replanted at a rate that exceeds the annual harvest. Thus, this species is a renewable resource that is indigenous to our country with a significant percentage of the supply growing on federal lands. Over the past 5 year period, the price increases on top quality (clear) Douglas fir have been considerably less than the inflationary rate. This in part is due to the fact that very low levels of energy are needed to turn the wood log into usable stock (veneer or dimensional boards). In comparison, many materials requiring high levels of energy to produce have increased at a much higher rate by percentage (table II).

At present, we are able to purchase select, clear, vertical grain 1/16 inch thick Douglas fir veneers for about 80¢ per pound ready to use (trimmed) in the mold, which is competitive with most of the other materials being considered for turbine blades. It is thought that the price of wood will look even more favorable in the near future as energy costs continue to increase.

Fabrication is, of course, the major cost factor when building wind turbine blades of any material. We feel that our costs to fabricate wood blades on a production basis can be very low. We have not yet worked out all of the details, but within the next two months we will be finalizing a manufacturing plan which will be discussed in a final report under our present NASA/DOE contract No. DEN3-101.

TABLE I. - RELATIVE EFFICIENCY OF VARIOUS MATERIALS  
IN DIFFERENT ROLES

Material	Young's modulus, E, MN/m <sup>2</sup>	Specific gravity, $\rho$ , g/cc	Simple tension & compr.	Column buckling	Panel buckling
Steel	210,000	7.8	25,000	190	7.5
Titanium	120,000	4.5	25,000	240	11.0
Aluminum	73,000	2.8	25,000	310	15.0
Magnesium	42,000	1.7	24,000	380	20.5
Glass	73,000	2.4	25,000	360	17.5
Brick	21,000	3.0	7,000	150	9.0
Concrete	15,000	2.5	6,000	160	10.0
Carbon-fibre composite	200,000	2.0	100,000	700	29.0
Wood (spruce)	14,000	.5	25,000	750	48.0

Taken from the book "Structures" by J. E. Gordon.

TABLE II. - ENERGY CONSIDERATIONS FOR WOOD AND OTHER MATERIALS

[Taken from the book "Structures" by J. E. Gordon.]

## (a) Approximate energies required for production

Material	Energy to manufacture, Joules $\times 10^9$ per ton	Oil equivalent, tons
Steel(mild)	60	1.5
Titanium	800	20
Aluminium	250	6
Glass	24	0.6
Brick	6	0.15
Concrete	4.0	0.1
Carbon-fibre composite	4,000	100
Wood(spruce)	1.0	0.025
Polyethylene	45	1.1

Note: All these values are very rough and no doubt controversial, but I think that they are in the right region. The value given for carbon-fibre composites is admittedly a guess, but it is a guess founded upon many years of experience in developing similar fibres.

## (b) Structural efficiency in terms of energy need

Material	Energy needed to ensure a given stiffness in the structure as a whole	Energy needed to produce a panel of given compressive strength
Steel	1	1
Titanium	13	9
Aluminium	4	2
Brick	0.4	0.1
Concrete	0.3	0.05
Wood	0.02	0.002
Carbon-fibre composite	17	17.0

(These figures are based on mild steel as unity. They are only very approximate.



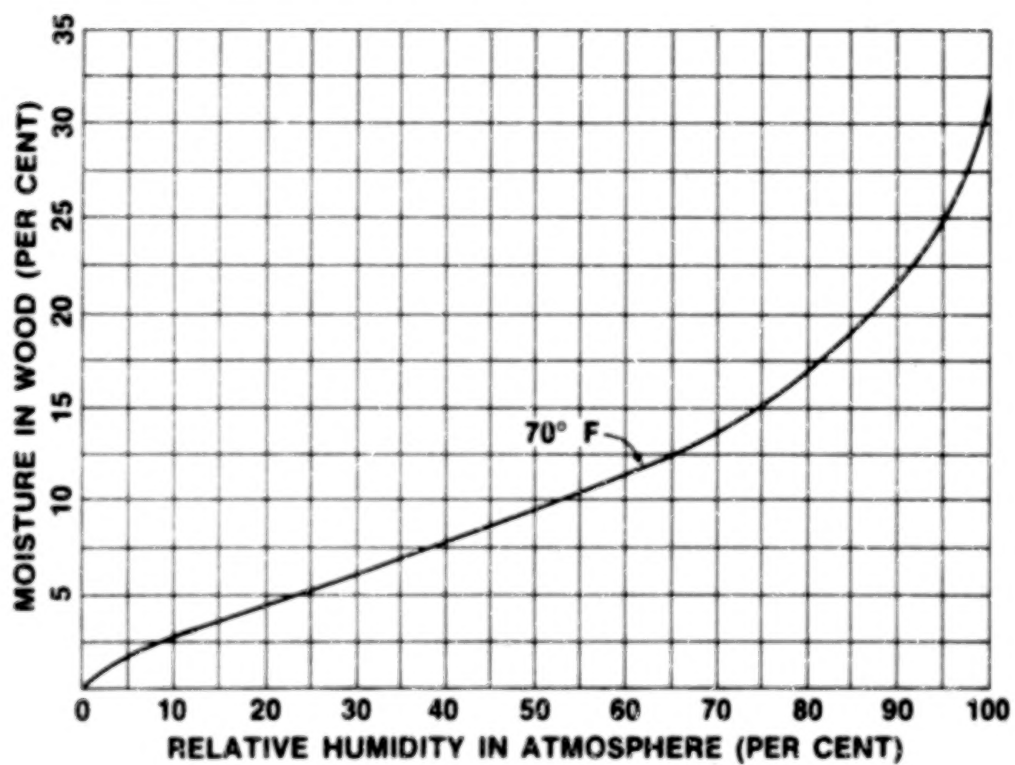


Figure 1. - Humidity chart.

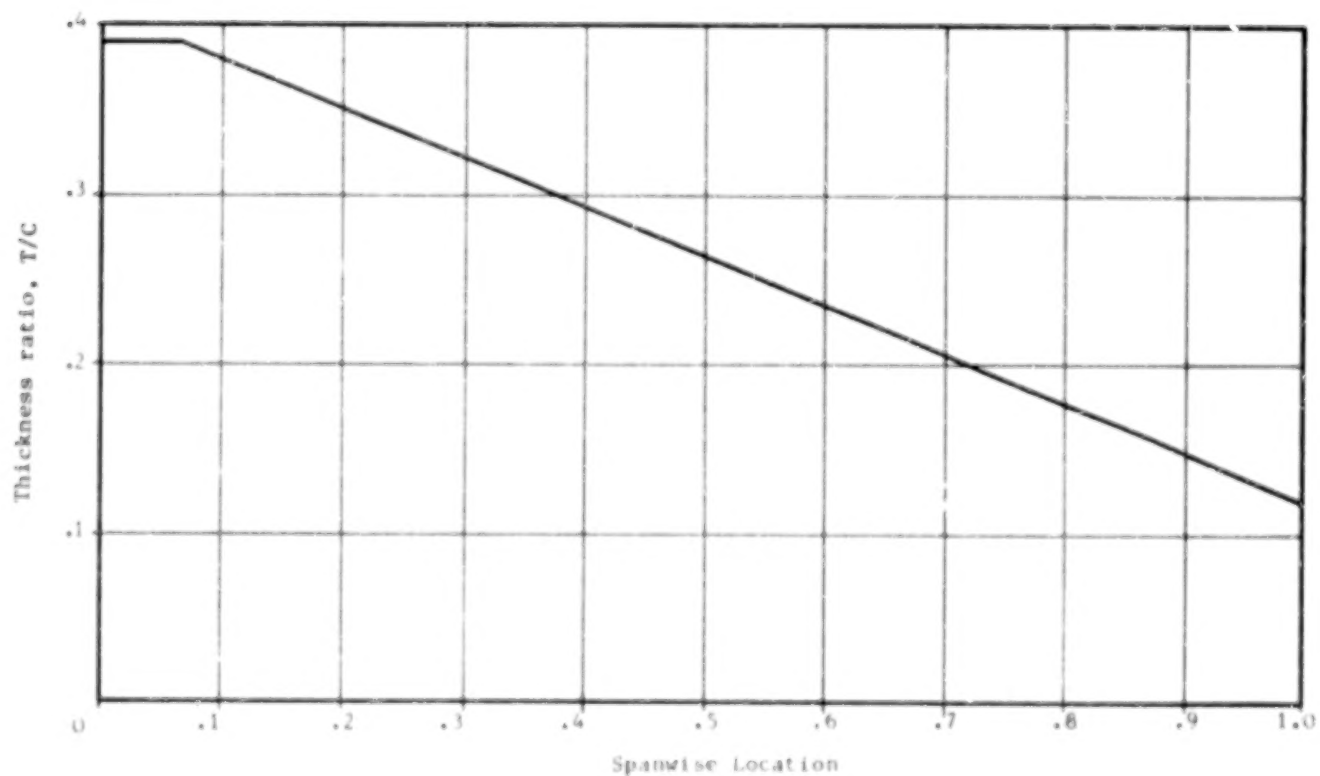


Figure 2. - Blade thickness variation.

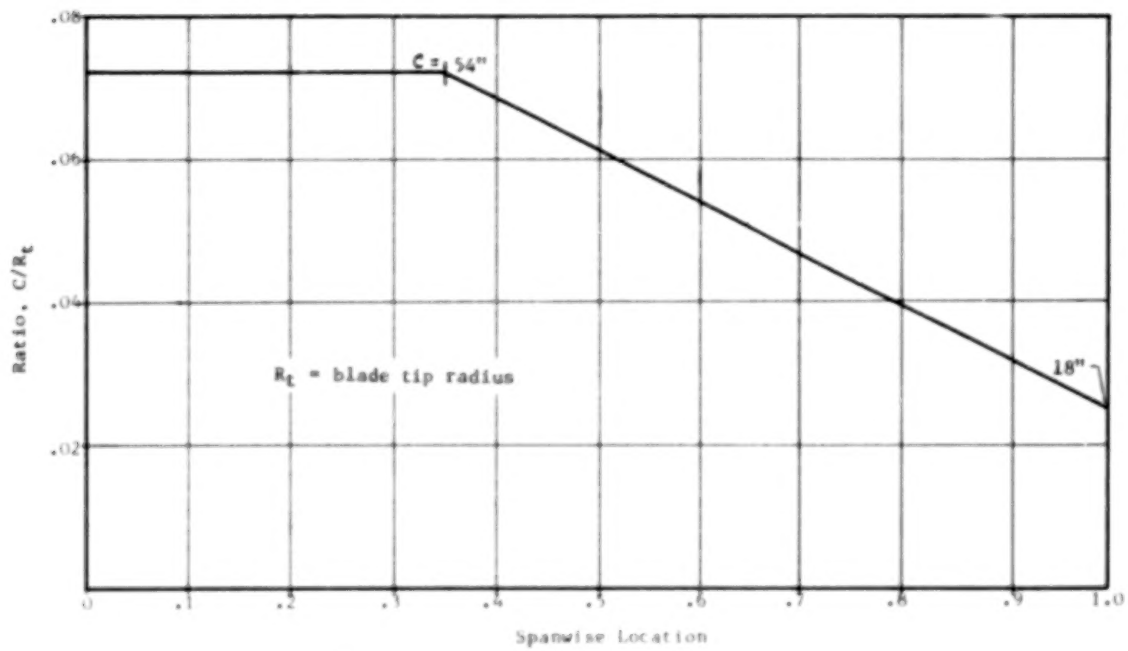


Figure 3. - Blade chord variation (plan form).

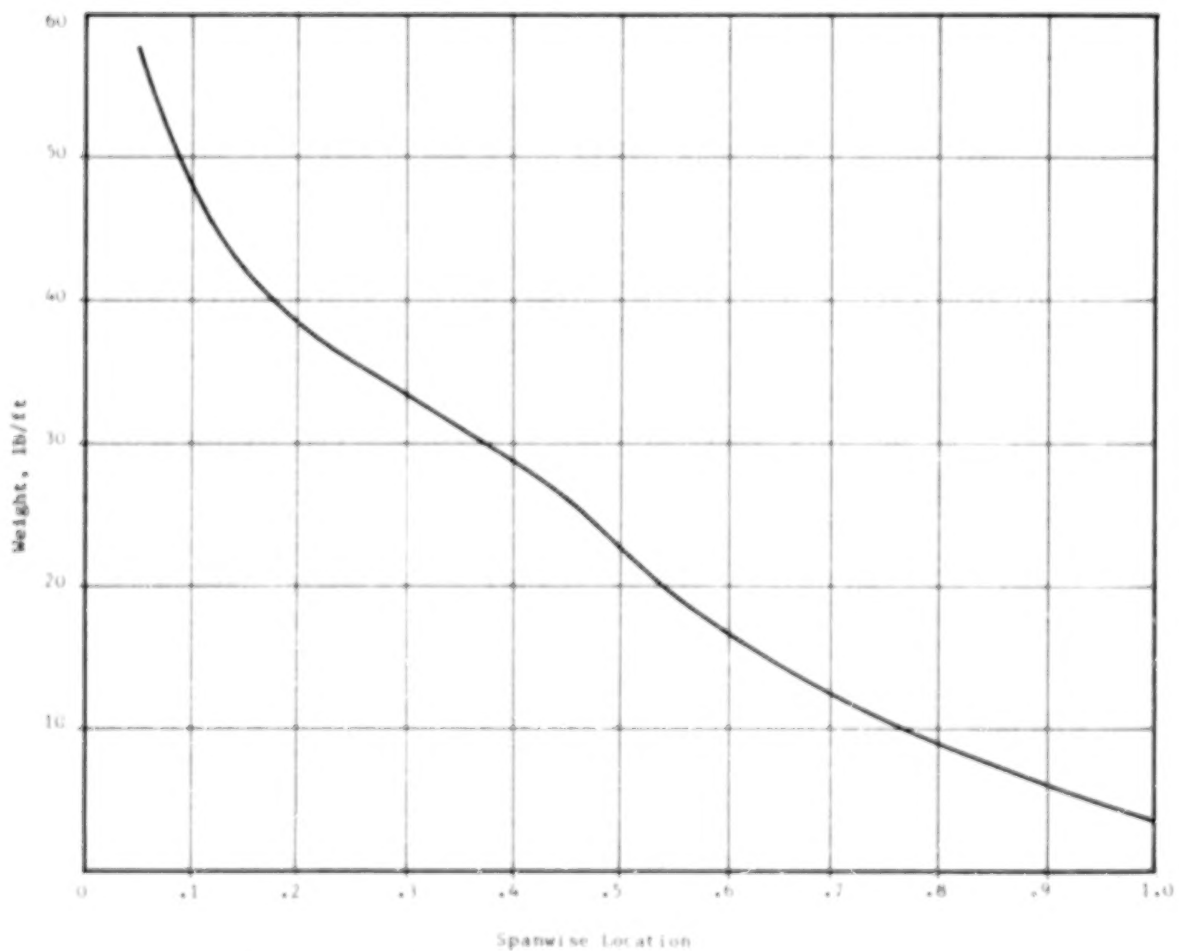


Figure 4. - Blade weight variation.

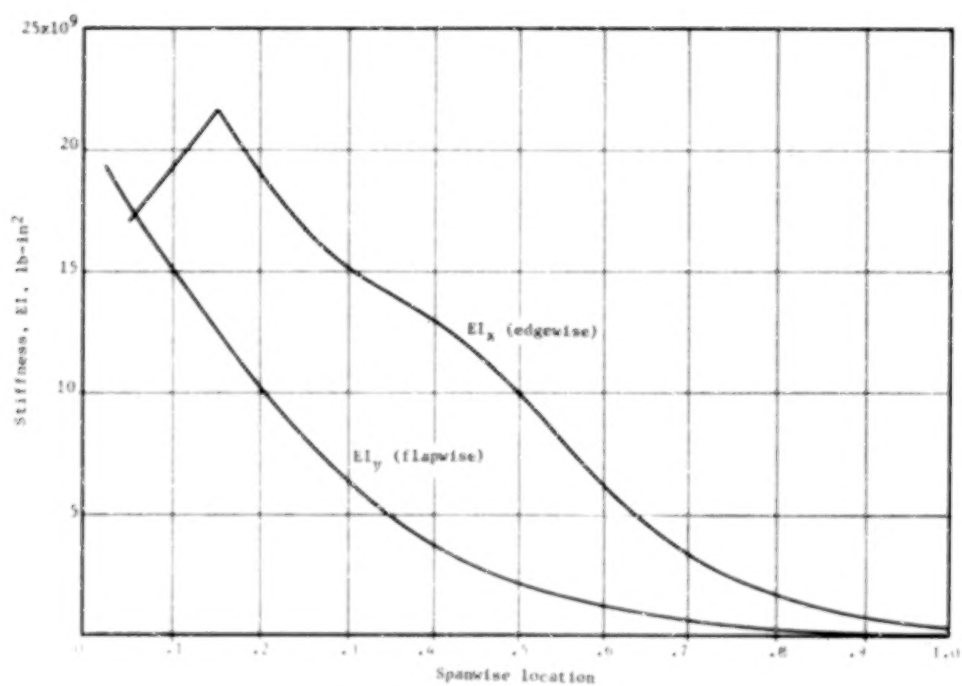


Figure 5. - Blade stiffness variation.

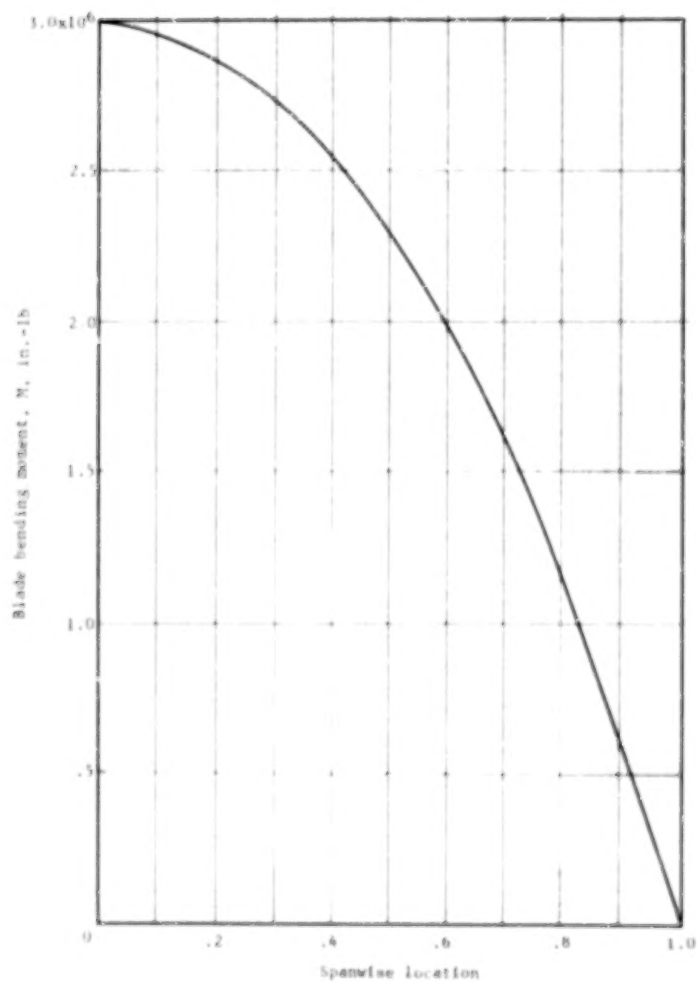


Figure 6. - Hurricane gust specification.

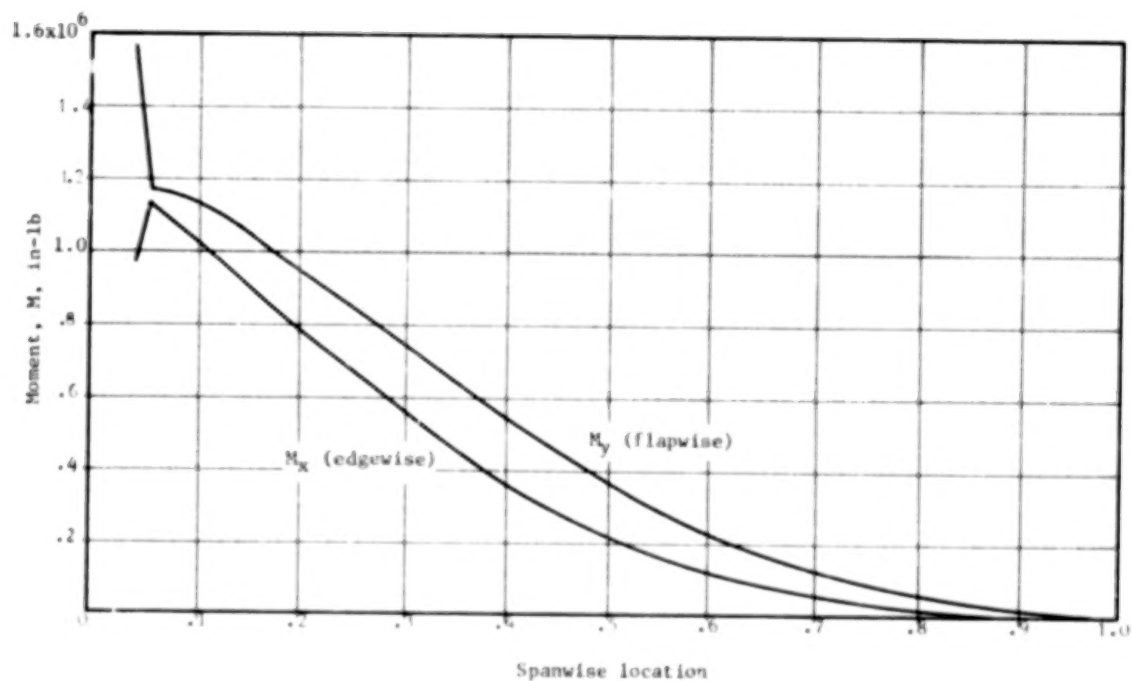


Figure 7. - Working load specification.

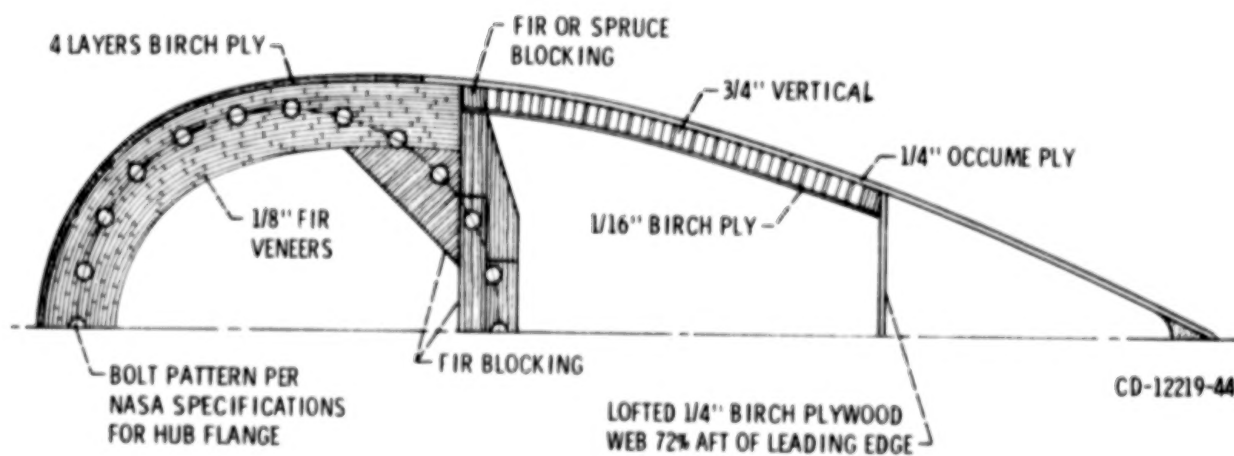
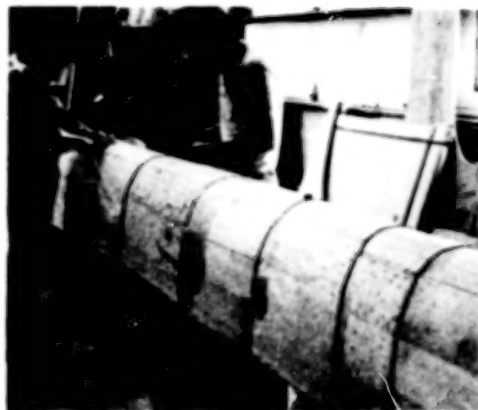


Figure 8. - Root end construction concept.



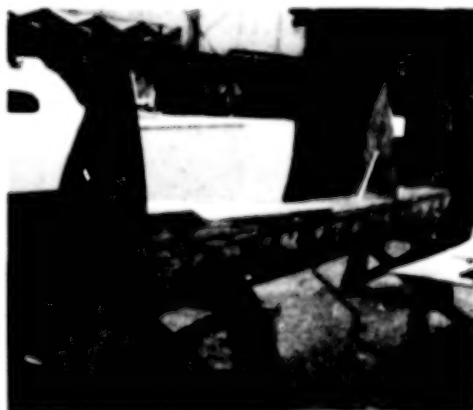
(a) Laminating the "D" section.



(c) Installation of compression strut and tail section on one side.



(e) Drilling pilot holes which will be enlarged to insert steel studs.



(b) Installation of sheer web and blocking at root end.



(d) Installation of opposing tail section and joining at aft tip.



(f) Studs inserted and ready to ship to NASA-LeRC for testing.

Figure 9. - Construction of 20-foot test specimen.

## LARGE, LOW COST COMPOSITE WIND TURBINE BLADES

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### INTRODUCTION

Various studies have shown that the cost of energy decreases with increasing rotor size in Wind Turbine Generator systems, and that the cost of the rotor is a major contributor to initial procurement and annual operating costs (References 1 and 2). In an effort to reduce rotor cost, NASA Lewis Research Center, with Department of Energy funding, initiated a program to develop a large, low cost wind turbine blade representative of a design for a 300 ft-diameter wind generator system. This paper describes the design, analysis, and test results of that program, and its extension to the follow-on program, fabrication of two composite blades for the Mod-1 200 ft-diameter wind turbine. Structural Composites Industries, Inc., Azusa, California, fabricated the spar for the 150 ft blade.

### 150 FT BLADE

Since the primary objective was fabrication of a large, low cost blade, the task was approached from the standpoint of selecting a commercially available low cost process and adapting the design to it. Among the several processes considered, including both metal and composite constructions, Kaman selected a composite design which employed a new application of a commercially available glass fiber material, recommended by SCI, which we named Transverse Filament Tape (TFT). TFT is a woven roving E-glass tape having all of its structural fibers oriented across the tape width. Use of TFT in the manufacturing process for the spar involved winding TFT onto a mandrel, with overlap, and simultaneously overwinding a layer of continuous filament rovings for compaction. Ninety percent of the material deposited was TFT, oriented along the spanwise axis of the spar. The overwound rovings (hoops) comprised the other 10 percent of material. Patent applications have been filed for certain aspects of this TFT process.

Special emphasis was placed on matching the design to the structural properties obtainable from the process, taking into account the anticipated commercial quality of the TFT laminate. Refinement of the process to typical aerospace standards was deliberately avoided. Determination of the material properties and structural capabilities of TFT were primary considerations in the 150 ft blade design and analysis effort.

### Design Description

Figure 1 illustrates the blade configuration which is essentially that originally proposed by Kaman and reported earlier as a design concept (Reference 3). Figure 2 illustrates the completed blade positioned for static tests.

Primary components are the TFT wound E-glass/epoxy spar, an E-glass/polyester trailing edge spline made from pultrusions, sandwich panels constructed of resin impregnated kraft paper honeycomb faced with glass cloth/epoxy skins, and a steel hub adapter. These components are joined by epoxy bonding, except for the hub adapter, which is mechanically fastened to the spar and spline. The total blade weight was 36,000 pounds; 23,000 pounds of composite structure and 13,000 pounds of steel adapter and hardware.

The spar is a D-shaped monocoque shell, tapered in planform, depth, and wall thickness to achieve desired bending stiffness, mass distribution, and aerodynamic shape. It has a 15 degree linear twist and is about six feet wide by four feet deep at the root, and two feet wide by seven inches deep at the tip. The wide spar at the root provides stiffness for edgewise tuning of natural frequency without requiring an excessively large trailing edge spline. The spar tip is narrowed to reduce outboard blade weight for flatwise tuning.

Spar wall thickness is 1.5 inches from root to midspan, and it tapers down to one inch at the tip. The nominal wall thickness is measured at the corners of the aft web, where laminate compaction is greatest. Thicker spar walls are evident where compaction is less.

Local reinforcement is provided at the inboard end of the spar for about three feet, by interleaving between courses of TFT a woven roving having a  $\pm 45$  degree bias orientation. This produces a four inch wall thickness of more nearly isotropic properties where the steel hub adapter is bolted to the composite spar.

Ten afterbody panels, five upper and five lower, are honeycomb sandwich construction of kraft paper core and glass skins. The panels range in length from 15 to 30 feet, and in weight from 144 to 433 pounds. Panel thickness varies from six inches at the root to two inches at the tip. Outer skins are two plies of 1583 glass cloth and inner skins are one ply. Local reinforcement is added at panel edges for attachment to adjacent structure. The 3/8-inch core is phenolic resin impregnated, and weighs 2.3 pounds per cubic foot. Sizing of panel thickness was dictated by the requirement to carry afterbody airloads and to stabilize the trailing edge spline under edgewise bending loads.

The trailing edge spline was fabricated by laminating E-glass/polyester pultruded planks with epoxy adhesive, and shaping to the desired contour. Steel cheek plates were bonded and bolted to the inboard end of the spline to transmit axial loads to the root end truss. The spline extends from the root to mid-span. A trailing edge closure of glass cloth extends from mid-span to the blade tip.

The composite subassemblies were joined by bonding with room-temperature curing paste epoxy adhesives. 35 psi bonding pressure was applied by pneumatic hoses retained in a steel framework. Prefabricated T-clips were fitted and bonded between the spar aft wall and the afterbody panel inner skins to improve the structural effectiveness of the panel inner skins.



Syntactic foam adhesive was injected into the cavity between the spar and the afterbody panel core to provide a shear connection between the afterbody panel and the spar.

The hub adapter was attached to the spar by 18 five-inch tapered bushings inserted into carefully machined holes in the composite. Each bushing was held in place with a three inch-diameter stud torqued to achieve a 400,000 pound preload which prevents the bushing from unseating on its loaded side. All machining cuts for each hole were made from a single setup at that hole, to achieve the alignment and squareness tolerances required for uniform load distribution in the composite.

#### Design Loads

Design of the 150 ft blade was based upon a downwind, 16 rpm rotor, and operating cases specified by NASA which provide representative critical conditions for the structure. The six cases are briefly identified as:

1. Rated power (1800 kW), rated wind (18 mph)
2. Increasing gust, 18 mph to 60 mph, plus 25 percent overspeed
3. Emergency feather in 11 seconds
4. Decreasing gust, 18 mph to zero mph
5. Hurricane wind (120 mph), non-rotating
6. Maximum yaw rate (2 deg/sec) at 50 mph wind velocity

(Wind velocities are at 30 ft reference height.)

Analysis of the five rotating cases revealed that Case 2 produced the highest fatigue loads for the spar. Although Case 2 was projected to occur only infrequently, Kaman conservatively considered Case 2 to occur continuously for design purposes, primarily because little is known about the frequency of occurrence of fatigue-producing loads in wind turbine systems operating for a number of years. Case 2, therefore, became the design driver for 30 year life requirements. Fatigue stresses in the spar associated with Case 2 loads were maintained below the estimated endurance limit of the composite material.

Case 2 was critical for both fatigue and static loads in the trailing edge outboard of Blade Station 18. Inboard of Station 18, Case 2 is critical for fatigue, and Case 6 is critical for static loads in the trailing edge spline and its attachment to the root end truss.

Case 5 produced the highest static loads in the spar, and was selected for static strength and buckling criteria. Case 5 loads were based on the conclusion that maximum aerodynamic force normal to the blade chord would be generated at the blade tip while the blades were feathered and parked horizontally. Although feathered, maximum force can be generated on the blade with only a 12 degree change in wind direction from the zero lift condition; therefore, the blade was designed for the maximum force case. The critical



orientation for Case 5 loads was a downward-acting force combined with gravity, which put the lower (flat) surface of the blade into compression.

Afterbody panels and their attachments to the spar and trailing edge were designed to Case 5 airloads, plus loads imposed by spar deflections.

#### Material Allowables

Transverse Filament Tape (TFT) has been used for many years in the manufacture of commercial, filament wound pipe. Small quantities (about 10 percent) have been added to pipe to improve axial strength and bending stiffness. In wind turbine blades, the percentage of TFT is much greater than in pipe; TFT comprises approximately 90 percent of the spar to provide much greater bending strength and stiffness. As a consequence of this primary structural duty, laminate characterization tests were conducted to provide material allowables for design.

Static characterizations were obtained via small specimen tests of TFT laminates. Thin laminates were laid up in the laboratory for tests at room temperature and 160°F, under both wet and dry conditions. Laminates having 20% and 35% resin content were tested. Hot-wet specimens were heat-soaked at 160°F and 95 percent relative humidity for 500 to 1000 hours before being tested within 15 minutes after removal from the environmental chamber.

Static properties obtained for 35 percent resin content, under the 160°F, wet conditions are shown below, along with the values used for design allowables derived from the hot-wet tests:

	<u>160°F, WET</u>	<u>DESIGN ALLOWABLES</u>
Ultimate tensile strength, ksi	52.7	33.7
Tensile modulus, $10^6$ psi	5.4	5.4
Ultimate compressive strength, ksi	44.2	41.4
Compressive modulus, $10^6$ psi	4.8	4.8
In-plane shear strength, ksi	3.46	3.16
Shear modulus, $10^6$ psi	0.305	0.305
Short beam shear strength (Interlaminar shear), ksi	3.32	3.12
Poisson's ratio	0.33	0.33

Strength properties of design allowables were reduced 3-sigma from the mean, whereas elastic properties were mean values.

Fatigue characterization was obtained from small specimen fatigue tests of sandwich beams having a TFT laminate on one side and a stainless steel sheet on the other, separated by aluminum honeycomb core. This configuration placed the neutral axis of the beam close to the stainless steel side, so that bending moments imposed on the beam resulted in primarily axial loads in the TFT laminate. The laminate was made with a TFT overlap in the

center, fully representative of the overlap obtained in the winding pattern for the spar. The objective of the fatigue tests was to determine whether there was a significant reduction in fatigue strength in the TFT structure when compared with a continuous-filament structure. TFT depends solely upon the resin matrix for tensile load transfer from one layer of glass rovings to an adjacent layer. The effect on fatigue strength of abruptly ending a roving layer across the primary stress direction was also of interest.

Fatigue testing these specimens proved to be a difficult task, involving many invalid failures as a consequence of specimen design. Initially, the TFT specimens were machined from flat laminate plates and then bonded to the sandwich beam for the bending fatigue tests. The machining operation produced cut fibers at the edges of the specimen which became failure loci producing invalid fatigue failures. Later, TFT laminates were molded to shape to avoid the cut edges of the machining operation. The molded specimens were better, but still produced invalid failures in the vicinity of retention grips. A better solution appears to be use of wound tubular specimens which eliminate laminate edges. Company-funded fatigue testing of tubular specimens has shown this approach to produce valid failures which provide better fatigue characterization than flat panel tests of composite laminate structures.

Results of the sandwich beam fatigue tests and a tubular specimen test are shown in Figure 3. The shape of the mean curve was based upon historical data from other sources and its location was based upon the sandwich beam tests. The data point for the single tubular specimen falls close to the mean curve, tending to validate the series. It is believed that the fatigue data presented in Figure 3 can be used with reasonable confidence that additional testing will not result in large changes in the position of the curve, and that it is unlikely that any such change would be toward lower values. To the degree that small specimen data are useful for design, it is believed that these data are conservative.

The mean curve was reduced three standard deviations (3-sigma) to provide a curve to be used for design. The allowable vibratory stress is obtained by applying the Goodman Diagram correction for steady stress using the 3-sigma-reduced fatigue endurance limit of 9000 + 7000 psi, and the 3-sigma-reduced ultimate stress of 48,900 psi for the 35% resin content, room temperature, dry condition.

In addition to the laminate characterizations described above, four quarter-scale specimens representing the blade root end attachment were fatigue tested to provide substantiation for the single-shear retention method. Double-ended specimens contained the same proportions of TFT, hoop rovings and + 45° bias tape as in the full-scale spar. Hardware details and installation procedures were also representative of the full-scale structure.

Specimens were tested in a tension-tension mode. Two were tested to 2 million cycles at normal operating loads, and two were tested to 10 million cycles. In an attempt to produce a failure, the last of the four specimens was tested at the Case 2 gust condition for 10 million cycles. Bearing

stress range in the bolt holes was 6500 - 19,400 psi during that test. No failures were produced in any of the specimens. It was concluded that the design values and interleaved laminate construction used for the root end composite structure were satisfactory for the full-scale spar.

Material allowables for the afterbody structure and its attachment in final assembly were based upon handbook data and industry practice for the well-established designs employed. As a check, several sub-element tests were run to verify the bond strengths obtained from the fabrication process proposed for the complete blade. These tests included measurement of skin strength and various bond line strengths listed below:

<u>TEST</u>	<u>NUMBER OF SPECIMENS</u>	<u>AVERAGE STRENGTH</u>	<u>REQUIRED STRENGTH</u>
Afterbody skins, tensile strength	4	51,000 psi	10,000 psi
Skin to core bond tensile strength	2	175 psi	10 psi
Afterbody skins bond adhesive lap shear	8	2,280 psi	550 psi
T-clip to spar attach- ment (detail) tensile strength	2	565 lb/in.	45 lb/in.
T-clip to spar attach- ment (subassembly) tensile strength	1	613 lb/in.	45 lb/in.
Afterbody panel to spar attachment shear strength of syntactic foam	2	117 psi (core failure)	32 psi

#### Blade Cost

The actual cost of fabricating the first prototype 150 ft blade was just over \$10/lb, exclusive of tooling and other non-recurring costs. That blade was made on one of a kind soft tooling, plywood forms for blade final assembly, and jury-rigged support fixtures for drilling the root end adapter holes. The 60 ft trailing edge spline was carved by hand. The blade spar was wound in four steps by SCI on a low cost steel mandrel which had a steadyrest at mid-span to minimize bending deflections and fatigue stresses.

Improvement of obvious limitations in efficiency in the above soft tooling includes a stiffer, smoother mandrel which would allow spar fabrication in one step instead of four, use of a fixture capable of machining all root end holes without repositioning the fixture support structure or the spar,

fabrication of the trailing edge spline as a molded detail to eliminate hand carving, and use of a final assembly fixture that positions subassembly details with less hand-fitting. Implementation of these improvements is projected to result in an average cost of the next ten blades at around \$7/pound, and the 200th blade at \$3/pound. These costs include the cost of an operational hub adapter, lightning protection, erosion protection, etc., not provided on the prototype 150 ft blade.

### Blade Tests

After blade fabrication was completed, the hub adapter was welded to a 30 ft long load reaction beam for static tests and natural frequency determinations. Static tests included measurement of blade edgewise and flatwise stiffness and deflections, proof-load tests to design limit load in edgewise and flatwise directions, and an ultimate failing load test in the flatwise direction.

Natural frequencies were determined by manually shaking the blade to reveal the low frequency fundamental bending modes, and by impact tests for higher bending modes and torsion.

Blade stiffness and deflection measurements were made by applying nominal loads at the blade tip, recording strain gage data along the blade, and measuring blade deflections from a reference line. Similar measurements were made during the limit load tests, and proved to yield better results with less data scatter.

The limit load test in the edgewise direction was based upon design loads under Case 6, the yaw condition, which is critical for the trailing edge structure. The limit load and ultimate failing load test in the flatwise direction was based on Case 5, the hurricane wind condition (164 mph wind at hub height), which is critical for spar buckling.

After completion of the natural frequency and stiffness determinations, and the edgewise test to design limit load, the blade was repositioned for the flatwise tests. The blade was tested to design limit load in the flatwise direction and then taken to failure at 9 percent above design limit load. Failure occurred as local crippling at a visible flaw in the spar laminate at blade station 45. Subsequent investigation revealed that the flaw was a local bulge in 60 percent of the spar wall thickness, resulting from the four step winding process and the associated soft tooling. Future blades will be made in a single step with improved tooling to eliminate such flaws.

Subsequent to the ultimate load test of the complete blade, the outboard 100 ft of blade was still structurally intact, so it was set up and tested in flatwise bending as a simply-supported overhanging beam. The test section from station 90 to 150 had none of the local flaw observed in the inboard region of the spar where the previous test had resulted in failure.

The outboard test section successfully sustained bending moments in excess of the ultimate design condition (defined as 1.5 times design limit) from

blade station 106 to the tip. At blade station 130, the applied moment was 2.8 times design limit, well above the predicted buckling strength of the spar wall.

This test demonstrated that large knockdown factors from theoretical crippling strength predictions are not necessary for pure monocoque glass/epoxy structures of this type, provided no serious material defects (such as the local bulges present at the station 45 failure location) are present.

#### MOD-1 BLADES

Kaman will fabricate two 100 ft composite blades for the Mod-1 wind turbine. A special challenge in the Mod-1 composite blade program is the requirement that the blades be designed to meet Mod-1 interface conditions established for the steel blades presently employed on the machine. Consequently, the composite blades must match steel blade weight, stiffnesses, deflections, frequencies, etc., to be compatible with the wind turbine system.

The blades will be designed using the technology developed under the 150 ft blade program described above, but with improvements clearly indicated by the results of that program. Also, the Mod-1 blades will be provided with lightning protection, leading edge erosion protection and paint.

The program will deliver two blades in 12 months, including proof testing. A prototype spar will be built and tested to ultimate load in flatwise bending to demonstrate freedom from buckling instability before spars for the two blades are built.

#### Design Description

Figure 4 illustrates the configuration of the Mod-1 composite blades. The spar is wider and deeper than on the 150 ft blade to match stiffness and tuning requirements of the Mod-1 machine. Additional material is added in the lower surface of the blade spar, as shown in the mid-span section view in Figure 5, to provide buckling stability for an emergency feather condition which includes 25 percent rotor overspeed and abrupt blade pitch change, causing large flatwise bending moments which deflect the blade towards the tower. The added material is unidirectional E-glass tape in which the structural fibers are oriented in the lengthwise direction of the tape (warp direction). This tape, termed Longitudinal Filament Tape (LFT), is laid down spanwise on the spar so that its structural fibers lie parallel to the structural fibers of TFT.

The afterbody panels of the Mod-1 blades are basically the same as on the 150 ft blade, the same materials and method of construction will be used. The trailing edge spline has been eliminated to save weight and reduce cost. This has been made possible by increasing the width of the spar for edgewise natural frequency tuning. The spar shape transitions from an airfoil shape at 40 percent radius to circular at the root where a conical steel adapter, shown in Figure 6, is bolted to the composite in a manner similar to the 150 ft blade. The adapter has an internal flange and bolt circle which interfaces with the Mod-1 rotor hub. The blade will be painted with an epoxy/polyurethane paint system and a leading edge neoprene



boot will be provided along the outboard one-third of the blade for erosion protection. A steel tip cap and braided wire trailing edge strap are planned for lightning protection.

#### Design Loads

Preliminary design loads for the Mod-1 composite blades are the bending moments predicted for the steel blades they will replace. Design loads will be updated by NASA Lewis as the composite blade design evolves. The highest flatwise and edgewise bending moments for static strength and buckling criteria are those which occur during the emergency feather condition shown in Figures 7 and 8. Peak loads during gust conditions and hurricane winds are lower than in the feathering case. It is required that the material proportional limit not be exceeded by these loads, which occur infrequently.

Bending moment distributions to be used for fatigue calculations are specified for 25 and 35 mph wind speeds, along with moment and frequency of occurrence spectra about the nominal values at those speeds. Moment distributions at 35 mph are shown in Figures 9 and 10. The blade is to be designed to withstand the cumulative effects of 435 million fatigue cycles during 30 years of operation when subjected to the 25 and 35 mph wind spectra.

#### Other Design Requirements

For compatibility with the existing Mod-1 wind turbine system, the composite blades must meet the following additional design requirements:

Maximum blade weight	20,000 pounds
Maximum blade tip deflection (emergency feather, overspeed condition is critical)	71 inches
Blade frequencies:	
1st Flatwise	1.17 - 1.45 Hz
1st Edgewise	2.80 - 2.98 Hz
1st Torsion	17.5 Hz

#### Material Allowables

The same composite material allowables developed for the 150 ft blade will be used for the Mod-1 blades, except that a 1/3 knockdown factor will be applied to elastic properties for buckling to account for fabrication variability. Therefore, the design ultimate load for buckling will be 2.25 times design limit load, whereas the design ultimate load for static strength will be 1.5 times design limit load.

#### Blade Tests

In addition to the ultimate load test to be performed on the prototype spar, both blades will be proof tested to design limit load in both flatwise and edgewise directions. Stiffness and deflection measurements will be made

edgewise directions. Stiffness and deflection measurements will be made during the proof tests. Natural frequencies will be determined for flatwise, edgewise and torsion modes.

#### CONCLUSIONS

- Design, analysis, fabrication and testing of a 150 ft composite blade have been successfully accomplished.
- Transverse Filament Tape (TFT) is capable of meeting structural design requirements for wind turbine blades
- Low cost fabrication of large wind turbine blades has been demonstrated
- Fatigue design allowables should be based on tests of wound specimens instead of flat laminates to minimize test terminations which obscure the true fatigue performance of composite structures
- Composite blades can be designed for interchangeability with steel blades in the Mod-1 Wind Generator System.

#### REFERENCES

1. Kaman Aerospace Corporation, "Design Study of Wind Turbines, 50 kW to 3000 kW for Electrical Utility Applications, Analysis and Design," NASA CR134937, February 1976.
2. General Electric Company, "Design Study of Wind Turbines, 50 kW to 3000 kW for Electric Utility Applications, Volume II, Analysis and Design," NASA CR134935, December 1976.
3. Sullivan, T. L., et al., "Wind Turbine Generator Rotor Blade Concepts with Low Cost Potential," NASA TM73835.

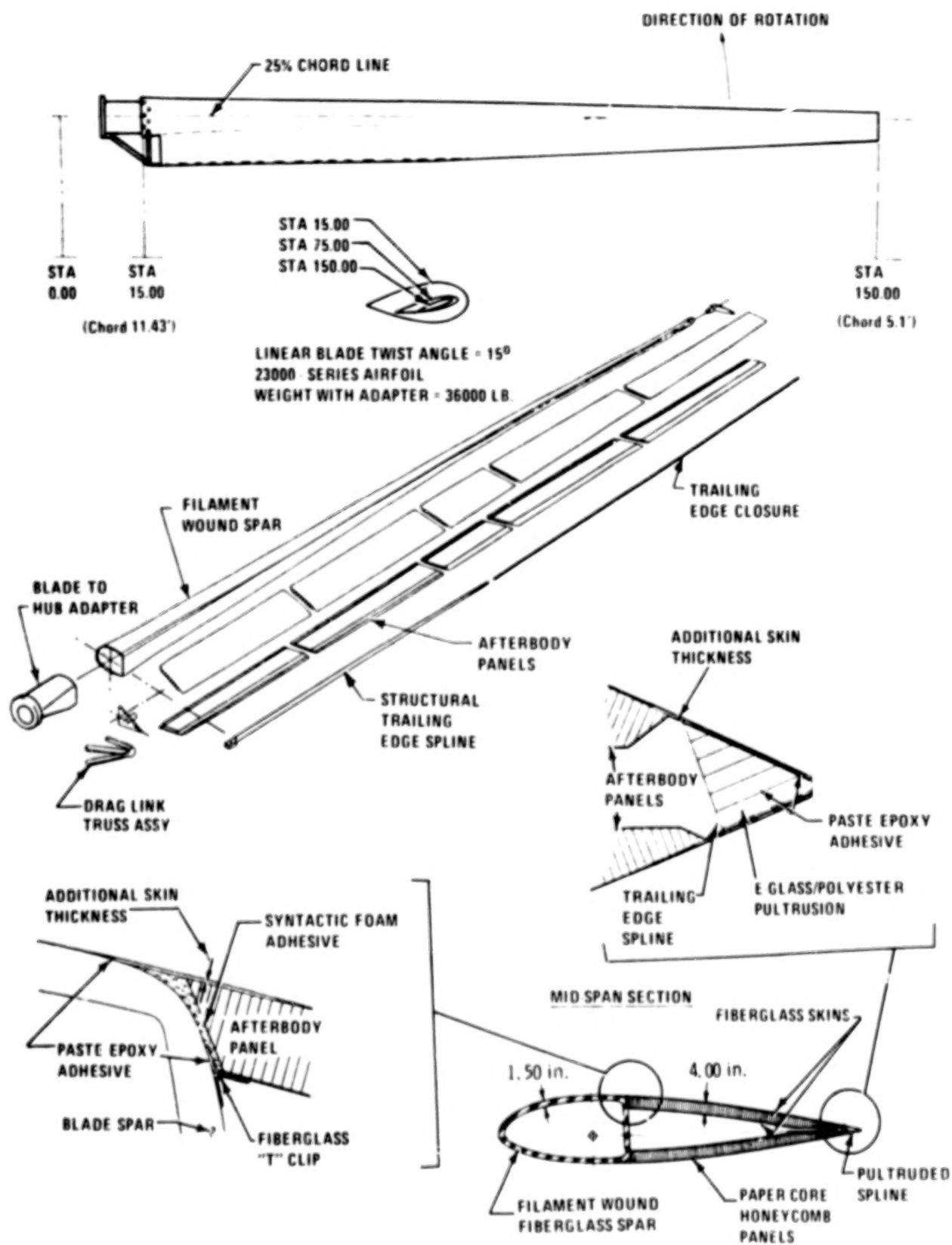


Figure 1. 150 Foot Wind Turbine Blade



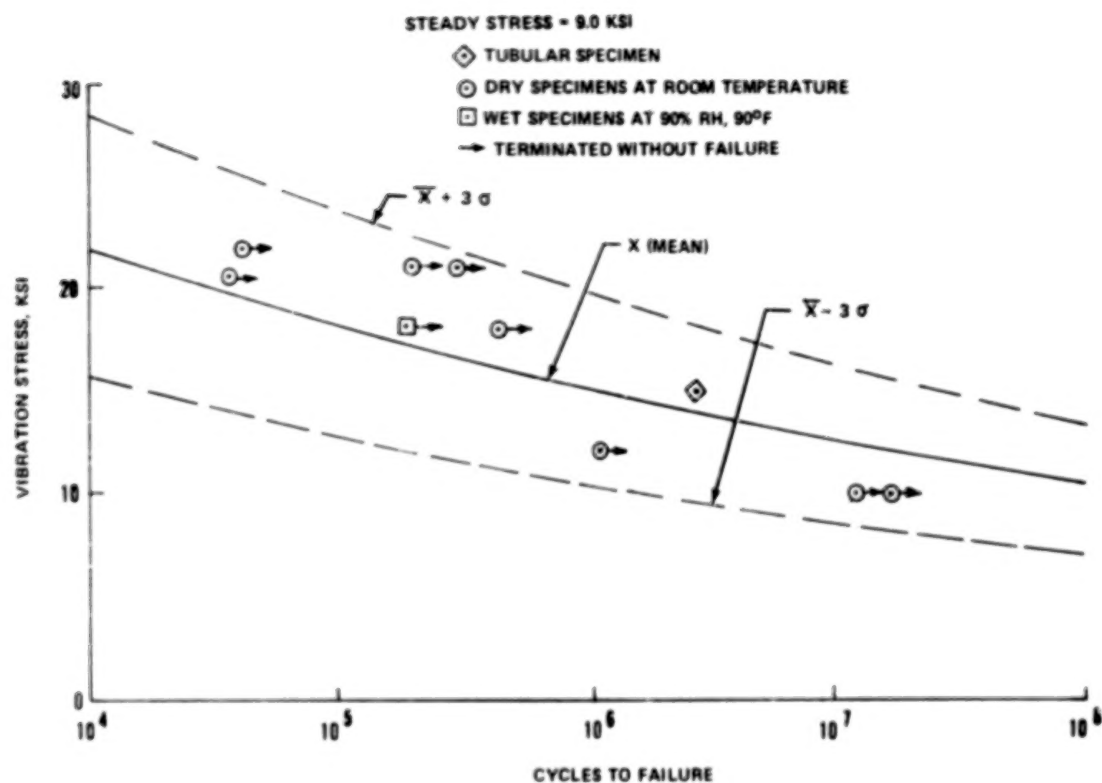


Figure 3. Spar Material Fatigue Characterization



Figure 2. Completed 150 Foot Wind Turbine Blade, Positioned for Edgewise Tests.

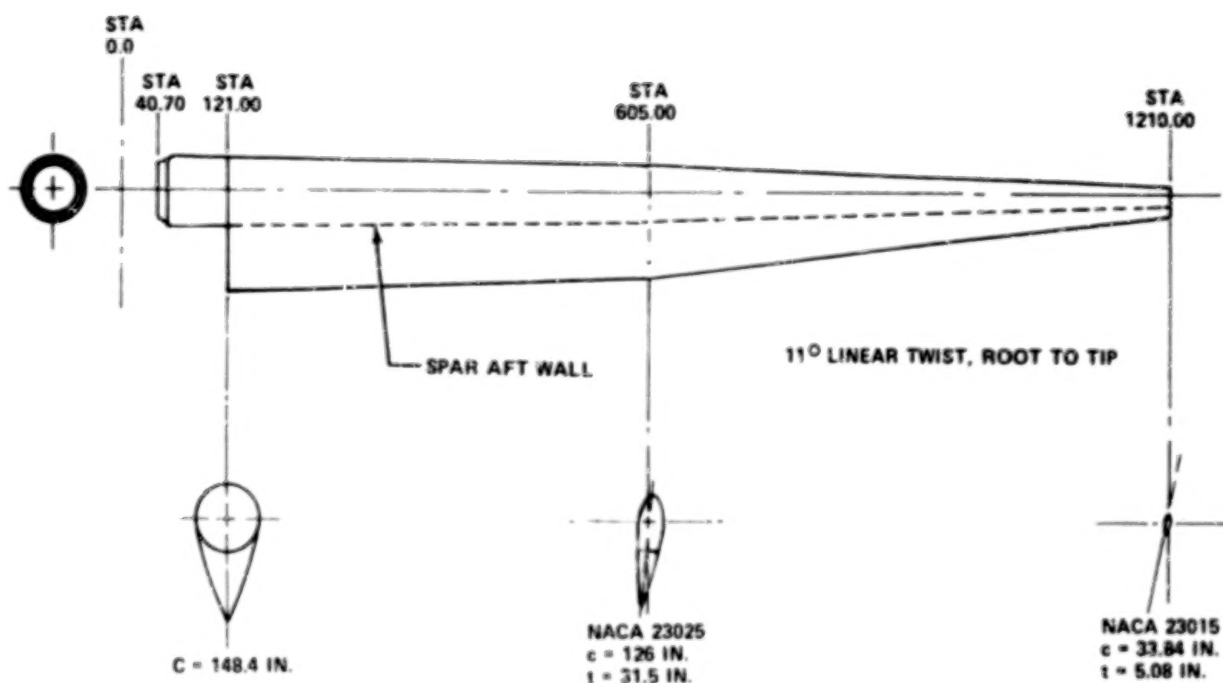


Figure 4. Mod-1 Composite Blade, General Arrangement.

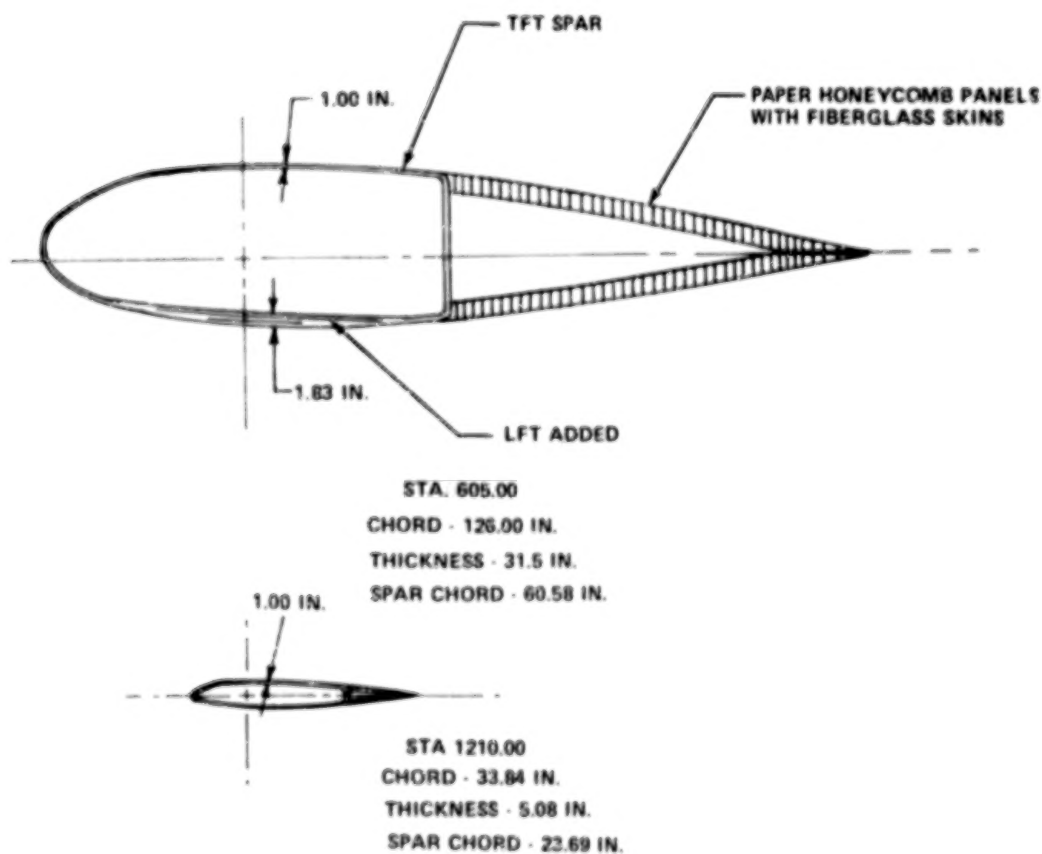


Figure 5. Mod-1 Composite Blade, Section Views.

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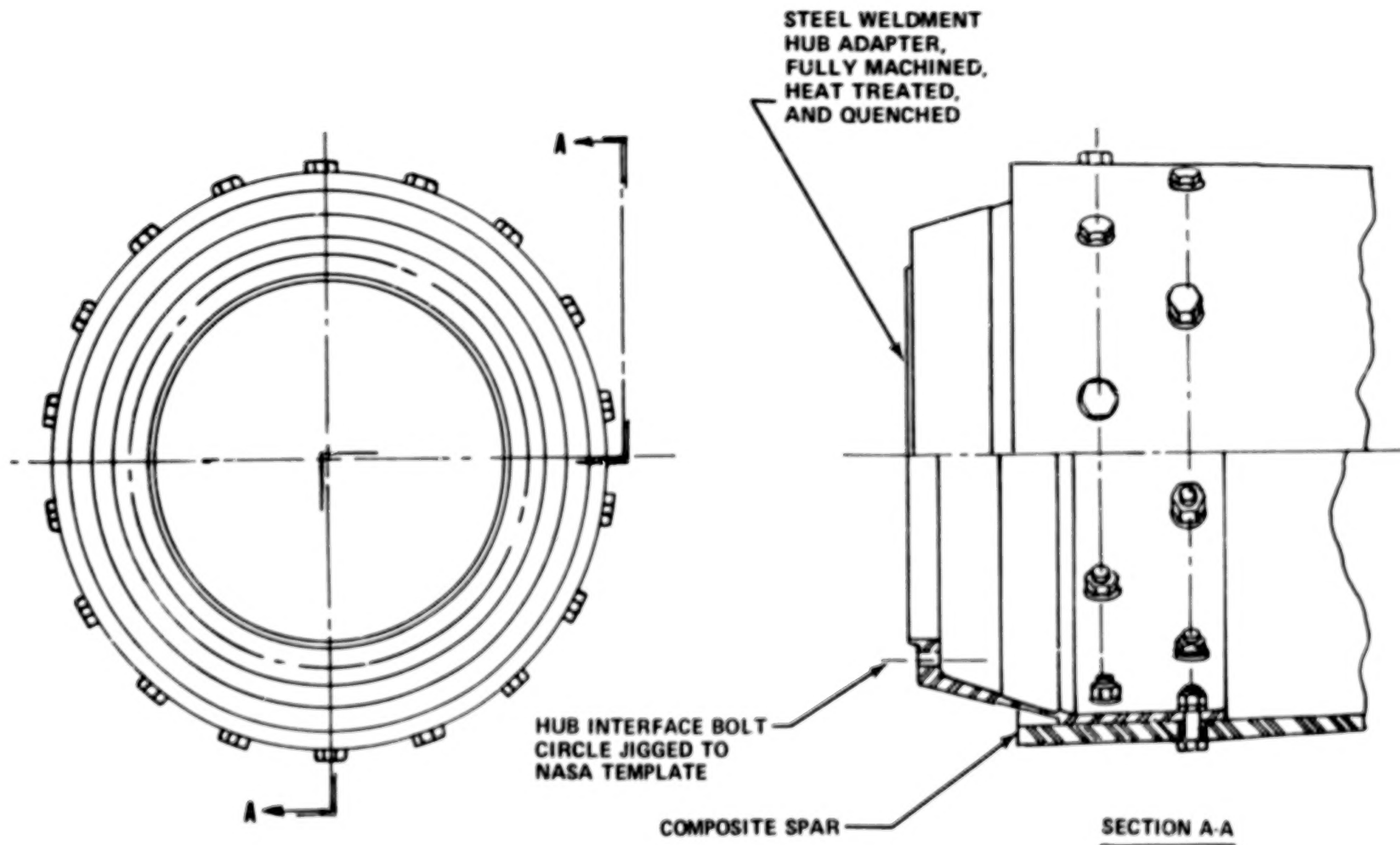


Figure 6. Mod-1 Composite Blade, Hub Adapter.

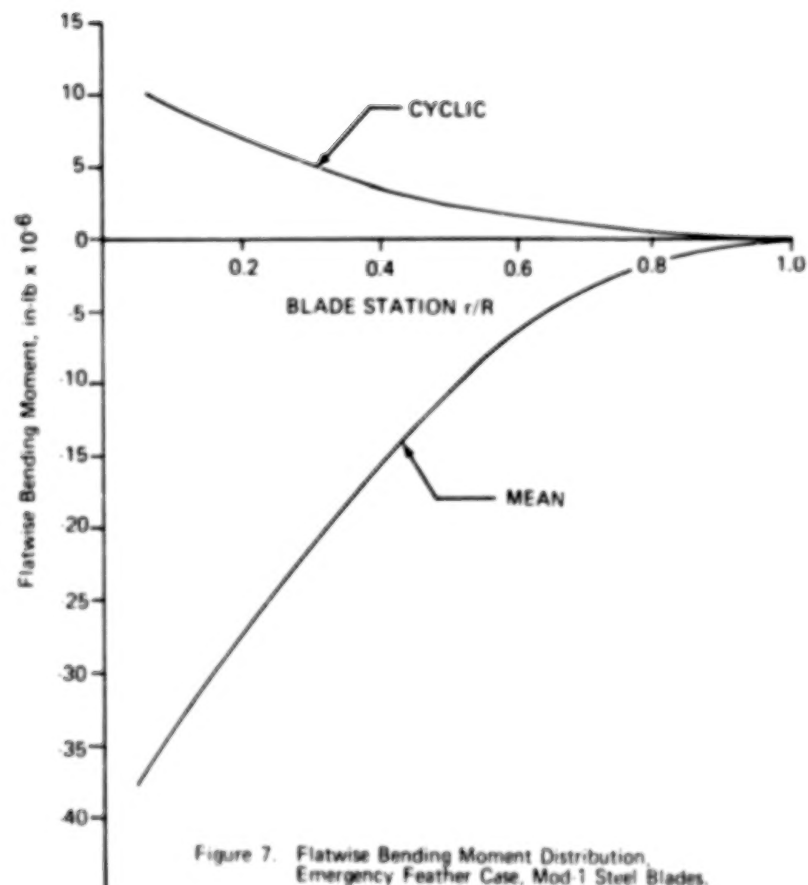


Figure 7. Flatwise Bending Moment Distribution, Emergency Feather Case, Mod-1 Steel Blades.

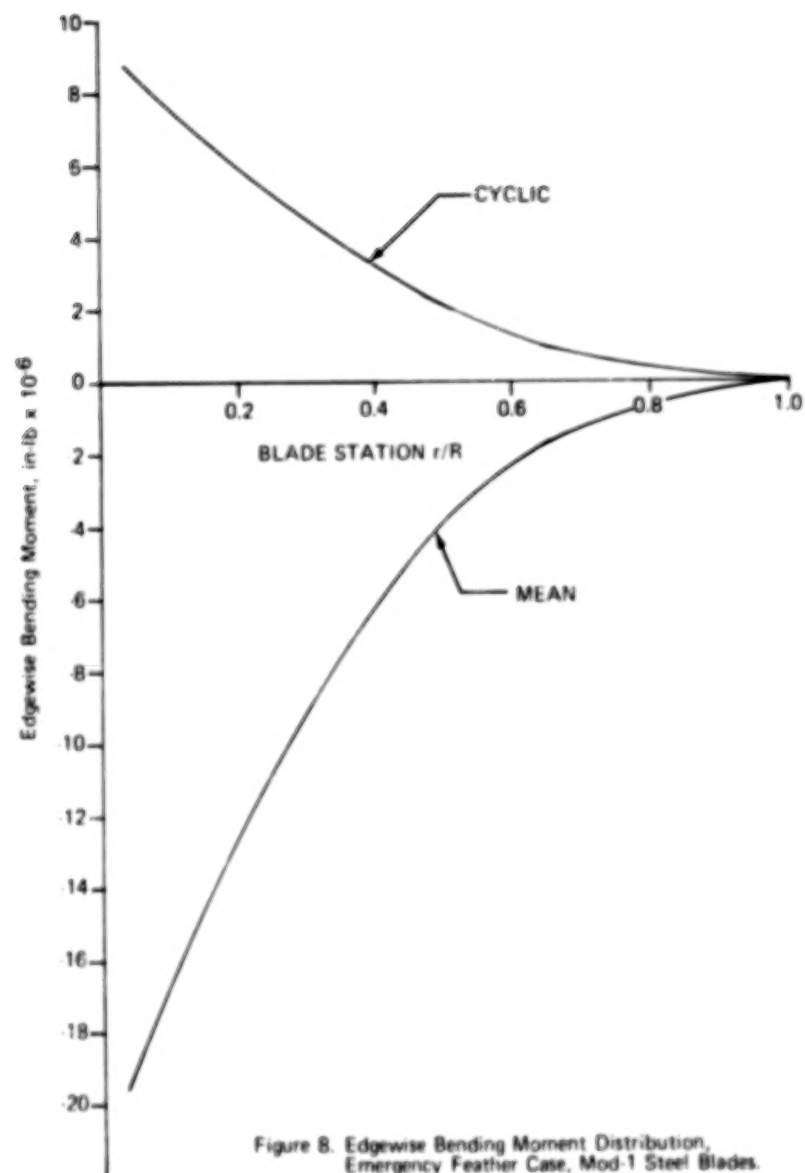
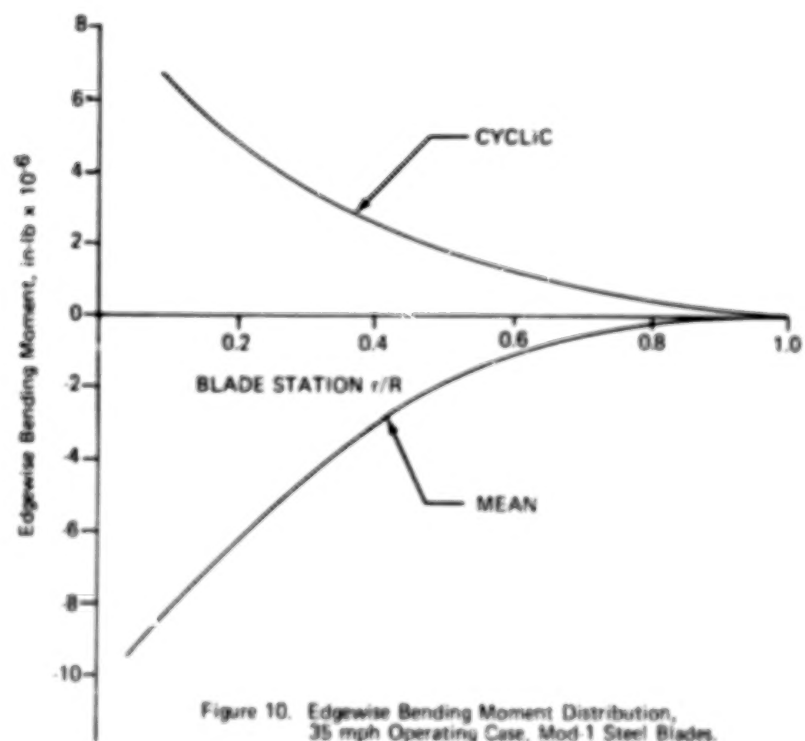
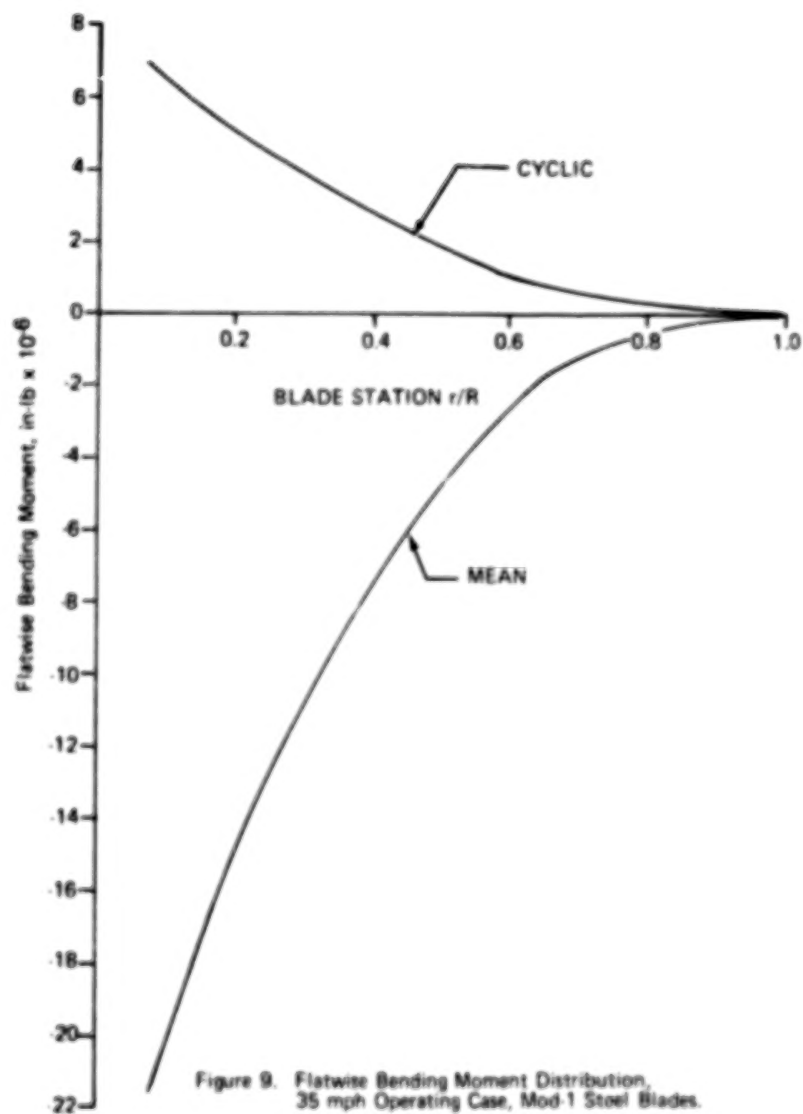


Figure 8. Edgewise Bending Moment Distribution, Emergency Feather Case, Mod-1 Steel Blades.



## THE MOD-1 STEEL BLADE

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### INTRODUCTION

Since September of 1977, design, development, fabrication, testing and transport of two 100 foot metal blades for the MOD-1 WTS has been completed. This paper summarizes that activity. Because the metal blade design was started late in the MOD-1 system development, many of the design requirements (allocations) were restrictive for the metal blade concept, particularly the maximum weight requirement. The unique design solutions required to achieve the weight goal resulted in a labor intensive (expensive) fabrication, particularly for a quantity of only two blades manufactured using minimal tooling. Nevertheless, the very existence of the blades represents a major achievement in large wind turbine system development.

### SPECIFICATIONS

The blade was designed to the GE Specification 273A6684, which also included an interface drawing 132D6479. The primary requirements are tabulated on Figure 1. For convenience, the requirements have been listed in geometric, structural, and performance categories, and the actual values achieved by the design have also been noted.

Weight control was a constant concern for this design. Fitting the blade structure to the specified weight limit required base metal fatigue allowables that would not allow use of mechanical fasteners and that required a better than "as rolled" surface finish.

### DESIGN DESCRIPTION

Each blade comprises a 97-1/2 foot long steel welded monocoque spar and a monolithic foam filled bonded trailing edge afterbody, as shown in Figures 2 and 3. Principal elements are (1) the spar, including the interface ring and the tip weight cavity; (2) the trailing edge (afterbody) structures; and (3) the joining system which attaches the T.E. to the spar. A detailed description of each of these elements follows:

Spar: A tapered, twisted, monocoque structure, formed in 15 foot sections of A533 Grade B, Class 2 material, and welded together. Upper surface plates are machined to provide "lands" for chordwise weld joints, as shown in Figure 4. The lower surface is stiffened with T-stiffener and frames for buckling resistance (see Figure 5). The hub flange is completely machined from a ring forging (A508) to efficiently use material to carry loads around the corner into the hub bolts (see Figure 6). Tip structure is machined from a block to provide leading edge radius (too sharp to be brake formed) with a cavity for incremental balance weights.

Trailing Edge: The six afterbodies are fabricated in 15 foot sections. Foam core blocks of different densities are bonded together and contoured to proper aerodynamic shape. Stainless steel skins (24 gage 301 1/2 hard) are bonded to upper and lower surfaces, and a cap is added at the extreme trailing edge. Conical lightening holes in the foam are included in the inboard sections only, as shown in Figure 7.

Joining System: The spar is prepared by construction of a flat interface surface using foam-in-place material with nominal 10 lb./cu. ft. density. The cured foam is surfaced and contoured. T.E. sections are bonded to the foam surface. Stainless steel (24 gage 301-1/2 hard) splice plates are installed across the chordwise joints between T.E. sections and along the spanwise joints. Butt joints in the splices are overlaid with similar gage cover plates. All exposed bond edges are covered with 2 inch wide 3 mil. stainless steel foil applied with polysulfide sealant for a moisture barrier. Stainless steel bands around the spar and trailing edge at approximately 5 foot intervals are also installed with the sealant material to provide a secondary attachment system.

#### DESIGN PROCEDURE

Blade design was accomplished in accord with established NASA design cycle and in close cooperation with GE who retained responsibility for all loads development and for system power. A three week trade off study with GE was the concept design phase and established the basic blade geometry that optimized the power within the constraints of our welded spar concept and the conditions of the contract.

#### DESIGN LOADS

Design load conditions were identified as frequent or infrequent, and were presented as integrated chordwise and flapwise

bending moment curves. For the frequent loading conditions, mean moments and cyclic moments were defined. A typical design curve is shown on Figure 8. Azimuthal phasing relationships of the chordwise to flapwise loads were supplied but for conservatism the design analyses combined the maximums. The critical frequent design condition, 35 mph wind velocity, 35 rpm, designed the structure for fatigue. Two infrequent conditions, emergency feather shutdown, a 38.9 rpm overspeed condition; and hurricane, 120 mph wind in the parked position, designed the structure for static buckling loads. The critical load diagram is shown on Figure 9.

Iteration of the design loads was accomplished by GE after the final design was completed. Based on final weight distribution and section properties, incremental loads were provided at 30° azimuthal angles for a finite element analysis. An ATLAS program was conducted with a total of 2,100 elements identified for the spar and 500 elements for the trailing edge structure. All calculated values showed positive margins for both the frequent and infrequent conditions. A typical stress distribution for the spar upper surface is shown on Figure 10.

#### ALLOWABLE STRESSES

Fatigue allowable stresses were established to be consistent with the AISC Handbook and the allowable developed by a fracture mechanics approach, considering the design spectra of wind loading conditions and the number of cycles expected at each wind velocity and gust factor. A simplified diagram of this approach is shown in Figure 11.

The selected allowable for RMS 125 surface finish base metal,  $S_r=28,600$  psi, was verified for both spar and trailing edge skin materials by a fatigue test program conducted using pre-cracked A533 specimens. This test program also established that welded metal has the same response as base metal. The loading spectra were not identical to the MOD-1 system, but were similar enough to provide verification. A more comprehensive description of this test is included in G. N. Davison's paper on the MOD-2 rotor.

AISC fatigue allowables for the various weld joint categories were validated by determining the limiting defect size that a fracture mechanics approach established and then providing the inspection techniques applicable to each joint configuration to discriminate defects smaller than this limit. The results of these analyses are shown on Figure 12.



Fatigue allowable (Sr) for the epoxy bonding system shown in Figure 13 was established at 240 psi. This value considered the maximum expected operating temperature and was based on data developed for helicopter blade repair. The value was further verified by a constant amplitude fatigue test program of three specimens at bond stress levels of 240, 158, and 75 psi, all of which completed  $5 \times 10^7$  load cycles without failure. Static allowables for bonded joints were established by a test program that tested lap shear specimens after exposure to various environments as shown in Figure 14. The trailing edge design stresses were verified by a fatigue test of a typical section which completed  $1.3 \times 10^7$  load cycles at 1.2 Hz and in various temperature and humidity environments without failure or evidence of bond deterioration. Figure 15 shows the test setup.

Buckling allowables for the spar were established by curved plate analyses based on Roark's theories and the Boeing Design Manual. The degree of difficulty in determining edge fixity and the overriding effect of initial panel straightness dictated a test program. A fifteen foot long spar specimen, including a chordwise weld joint, was fabricated using prototype tooling to control distortion. Initial tests resulted in premature failure. A longitudinal stiffener was incorporated at the center (25% chord) of the lower (compression) surface and the specimen sustained bending moment in excess of design ultimate load without failure (Figure 16).

Buckling allowables for the trailing edge stainless steel skin supported by the foam were determined using the Boeing Design Manual. Compression and shear moduli for the various foam densities were obtained initially from vendor data and verified by test of each foam shipment. In addition, compression tests of single-face sandwich specimens were conducted to validate the buckling stability of the T.E. sections with the lightening holes. Figures 17 and 17(a) show typical allowables and design stresses.

#### DESIGN FACTORS

The one overriding design factor that influenced many of the design decisions was the specified weight limit. The limit was just too low to allow alternate design solutions that might have resulted in a simpler design. Figure 18 summarizes the influence of the weight limit.

The overall size of the blade exceeded any known facilities for high temperature autoclave bonding. As a result, the

the room temperature epoxy system was developed for steel-to-steel and steel-to-foam applications. Also, local heating techniques for postweld stress relief of the spar weldment were required because available furnaces were too short.

While not specifically a design factor, the program schedule requirements influenced many of the design decisions. The spar section length (15 ft.) was selected to fit the capacity of a number of existing brake presses. Similarly, the decision to use stainless steel for the trailing edge skins was dictated when high strength carbon steel (4130) was not available in the required gage to meet the schedule requirements.

#### BLADE COST

The MOD-1 steel blade is without doubt a costly "Cadillac" structure. Even with the significant development and tooling costs not included, the costs exceeded \$40.00 per pound of structure.

The cost drivers were primarily the labor costs associated with fabricating a total of only two units to a tight schedule which excluded use of automatic production type tooling and processes.

Many of these experienced costs would be substantially reduced for fabrication of follow on blades. In addition, a different schedule could provide opportunity for material substitutions to reduce costs.

To be cost competitive, however, it appears that a significant investment in production tooling (and facilities) and an increase in the blade weight ( $\leq 25\%$ ) to eliminate machining, compression surface stiffeners and grinding will be required.

#### MAJOR PROBLEMS

The design was completed in six months (Final Design Review on March 15, 1978) and there have been no significant redesigns during the fabrication. During fabrication a number of problems occurred, as expected in a development program of this kind. The first occurred when the spar material we selected (A533 Grd B Cl 2) was bid by only one mill and required special mill run production. This delayed delivery and gave us a late start on fabrication of the spar.

A second setback occurred during in-place postweld heat treat of the first spar lower surface "clamshell" weldment. Severe distortion resulted from thermal gradients caused by improper heating techniques. See Figure 19. Although the weldment was almost completely flame straightened, engineering analyses would not confirm that full structural capability had been restored, and the weldment was replaced.

During final assembly, several small splice plates disbonded under no load conditions. Failure investigation established that the primer was not fully cured and that the final rinse prior to priming was inadequate, leaving a detergent film on the stainless and preventing the primer from adhering. Improved process control was established to prevent future occurrence, and the completed assemblies were mechanically tested to verify the bonding.

#### CONCLUSIONS & RECOMMENDATIONS

It is difficult and expensive to produce a blade structure to fit a set of predetermined constraints. Overall system trades earlier in the design process will reduce the downstream problems.

Fabrication costs can be reduced by minimizing the hand work requirements through design, tooling, facilities and mass production.

It is recommended that funding and schedules for this type of development program have an adequate reserve to allow resolution of unforeseen, unscheduled, and unfunded problems.

	Spec Requirements	Actual
<b>GEOMETRY</b>		
Interface to hub	56-1.25 inch dia. holes	Same
Length	97.5 feet	97.4 feet
Airfoil shape	NASA 44xx	Same
Twist	11° root to tip	Same
<b>PERFORMANCE</b>		
Operational life	30 yrs. ( $4.35 \times 10^8$ cycles)	To be determined
Design loads		
Frequent	35 mph wind (35 rpm) 24.8 mph wind (35 rpm)	Same Same
Infrequent	120 mph hurricane (static)	Same
	Emergency feather-overspeed (38.9 rpm)	Same
Balance weights (tuning)	500 lbs @ 40 lb. increments	516 lbs. (43 lb. increments)

	Spec Requirements	Actual
<b>STRUCTURAL</b>		
Material	Metal	A533 spar, 301 trailing edge
Weight	20,000 lbs. $\pm$ 1%	20,850 lbs - No. 001 20,710 lbs - No. 002
Frequency (Rigid mount)		
Flapwise	1.17 1.45 Hz	1.45 Hz (300 lbs. bal. wt.)
Chordwise	2.80 2.98 Hz	2.67 Hz (300 lbs. bal. wt.)
Torsion	>17.5 Hz	29.24 Hz (300 lbs. bal. wt.)
c. g. location	<35% chord aft of l.e.	33.66% aft - No. 001 34.19% aft - No. 002
Fatigue allowables		
Base metal - Cat. A (125 rms)	Sr=28,600 psi	Same
Welds - Cat. B	Sr=16,000 psi	Same
Cat. C	Sr=12,000 psi	Same
Cat. E	Sr=5,000 psi	Same

Figure 1. MOD-1 Primary Blade Requirements

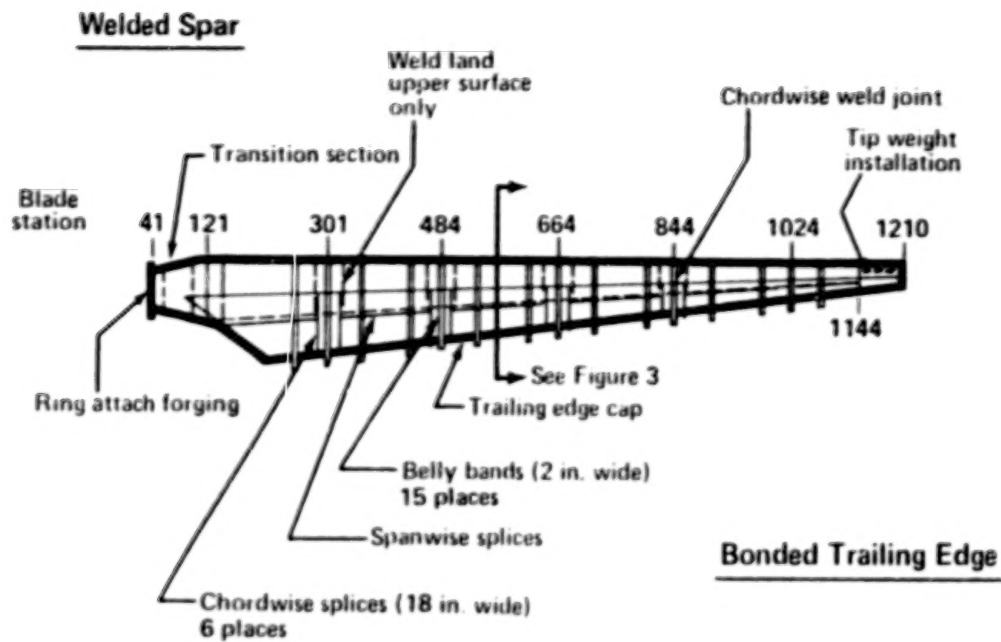


Figure 2. MOD-1 Blade Assembly

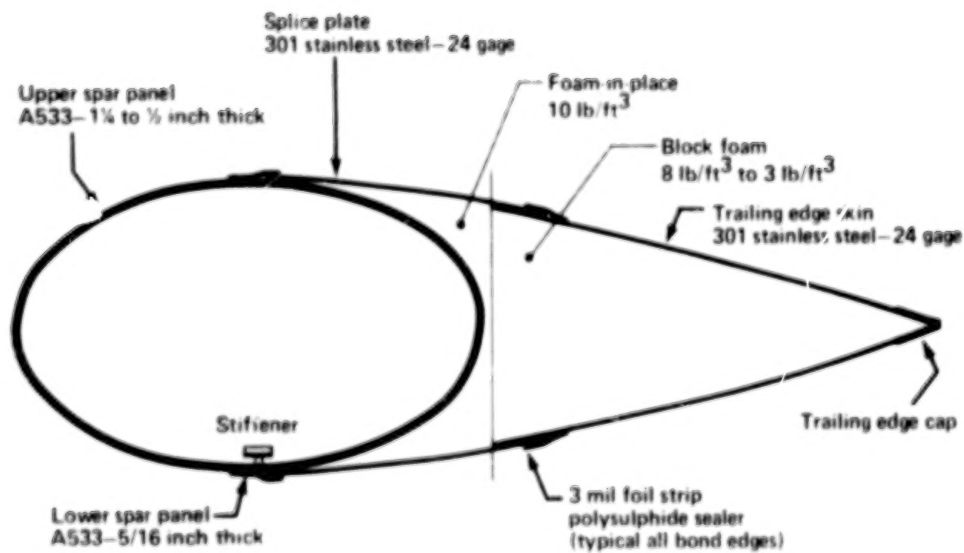


Figure 3. MOD-1 Blade Cross Section

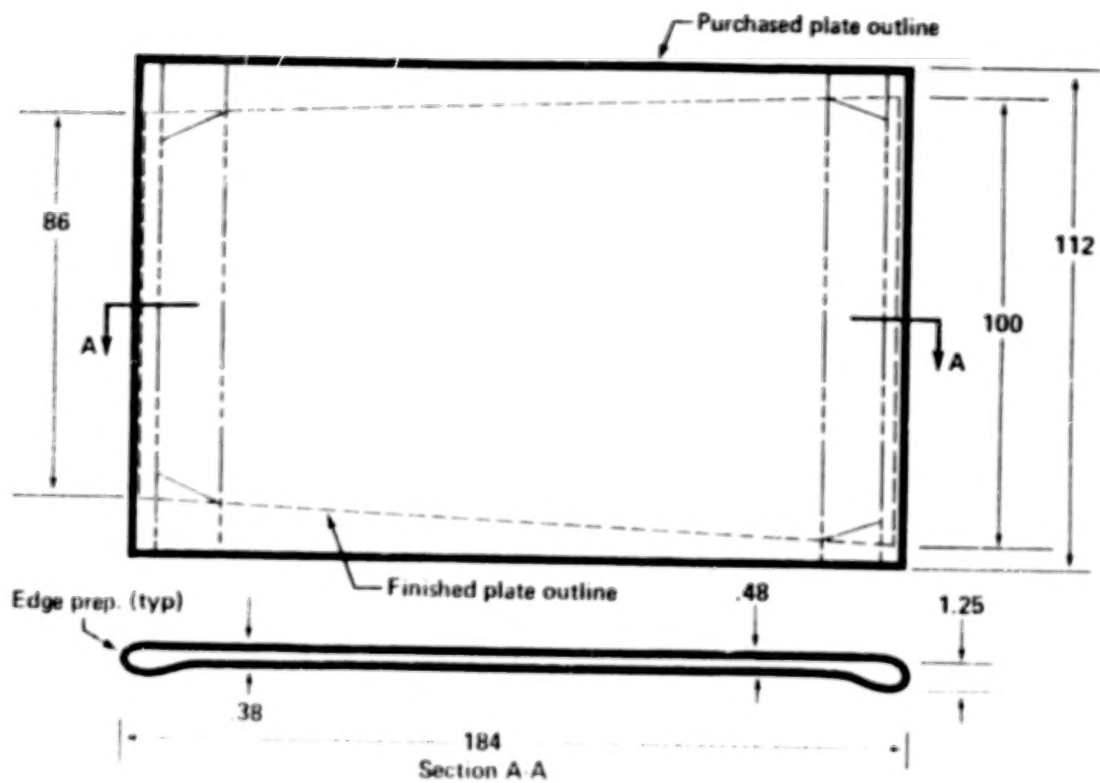


Figure 4. Upper Surface Plate (Typical)

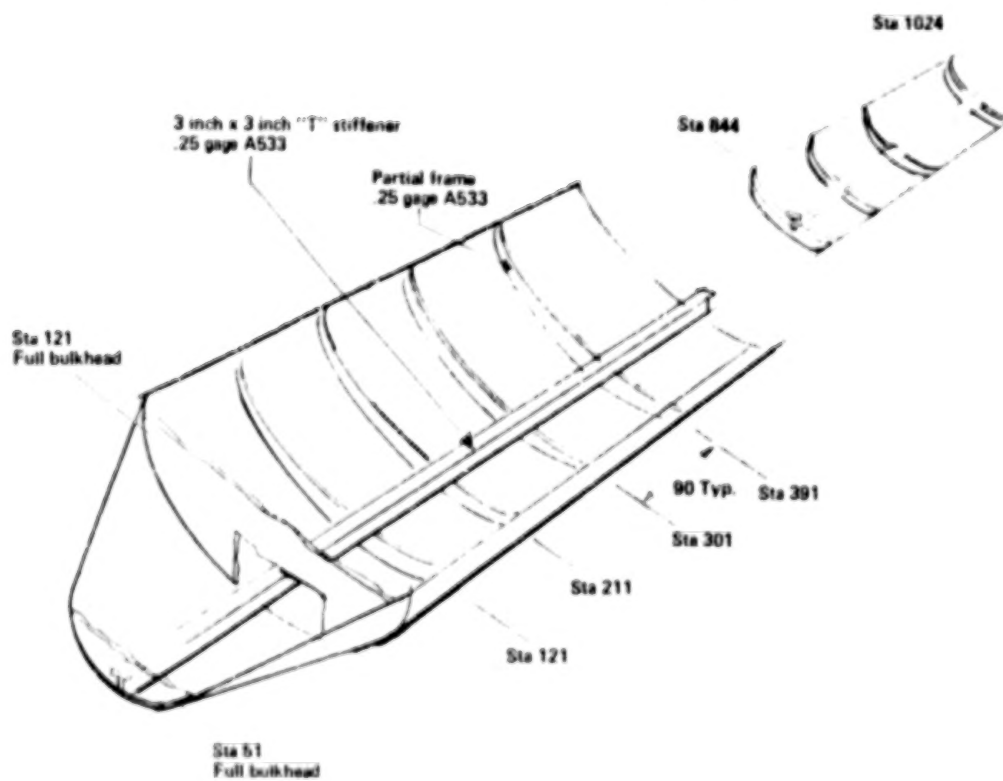


Figure 5. Lower Surface Stiffener

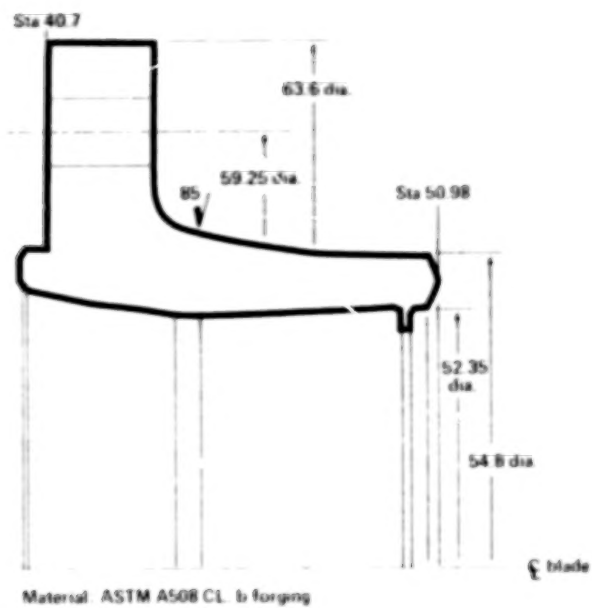


Figure 6. MOD-1 Blade Attach Flange

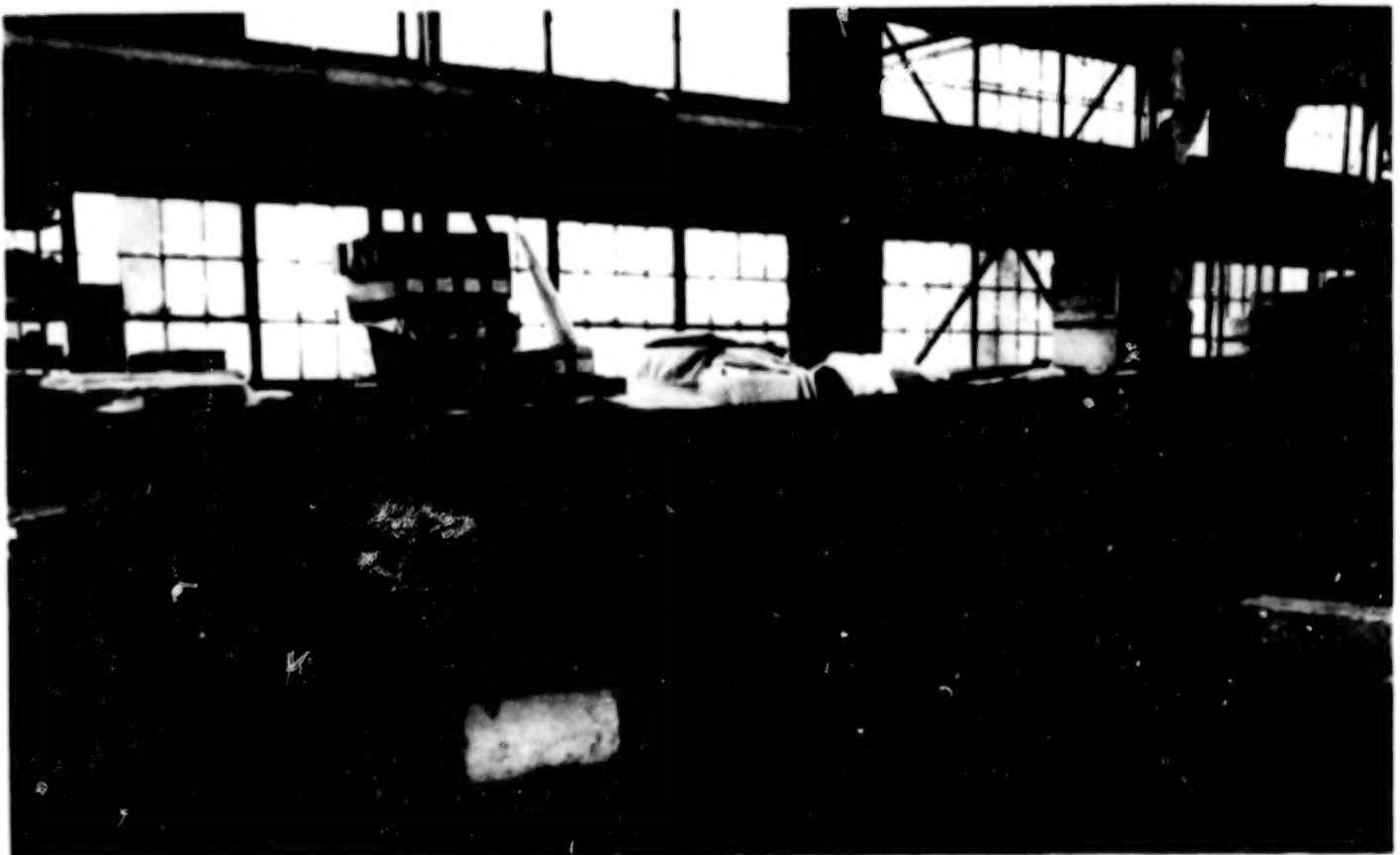


Figure 7. Typical TE Section.

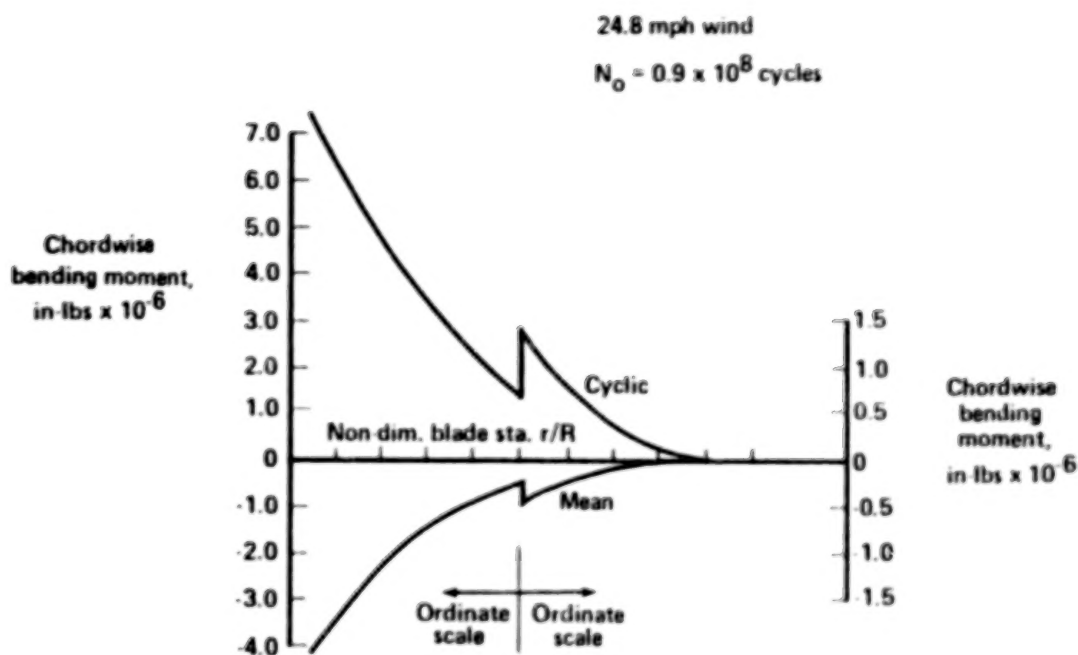


Figure 8. Chordwise Bending Moment Distribution

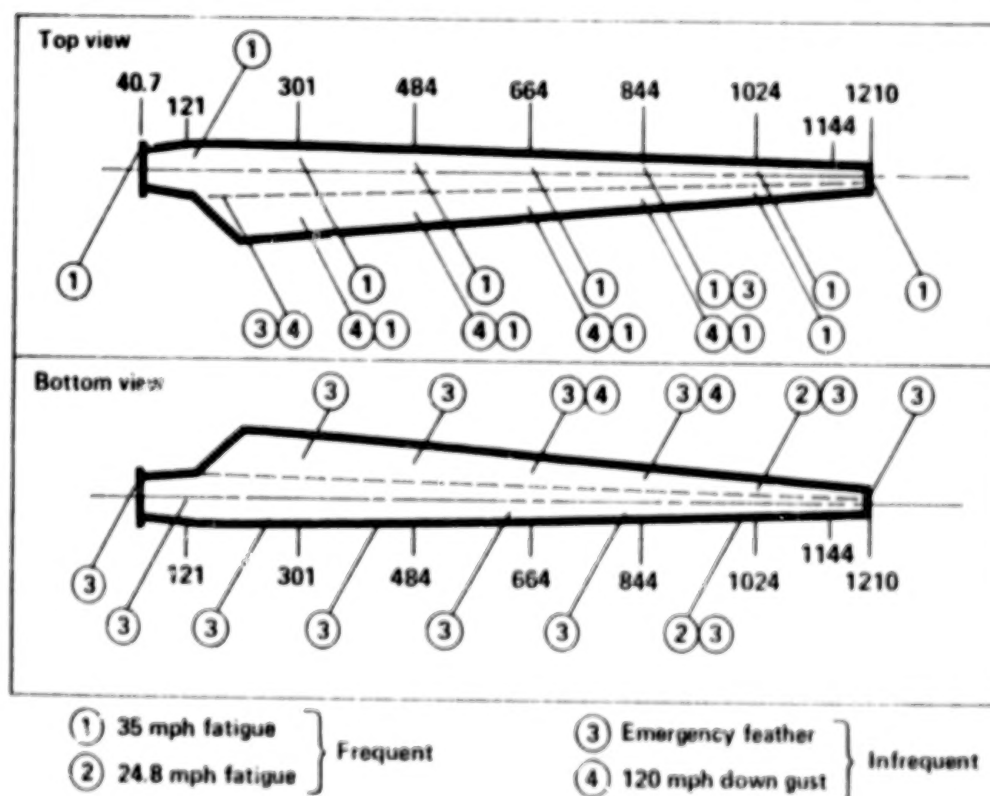


Figure 9. Design Load Conditions



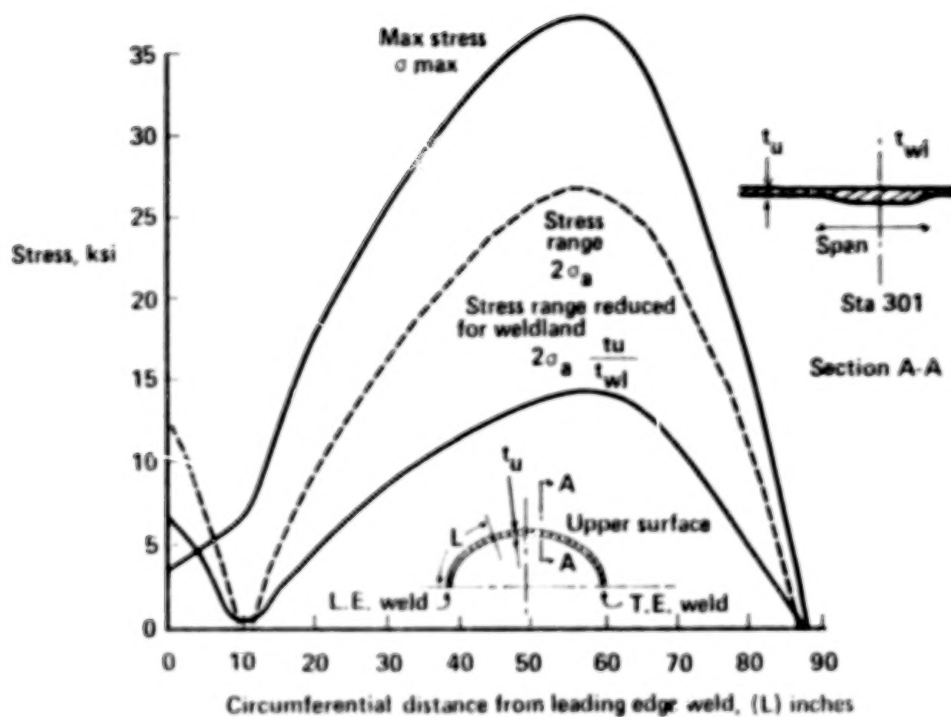


Figure 10. Upper Surface Stress Station 301

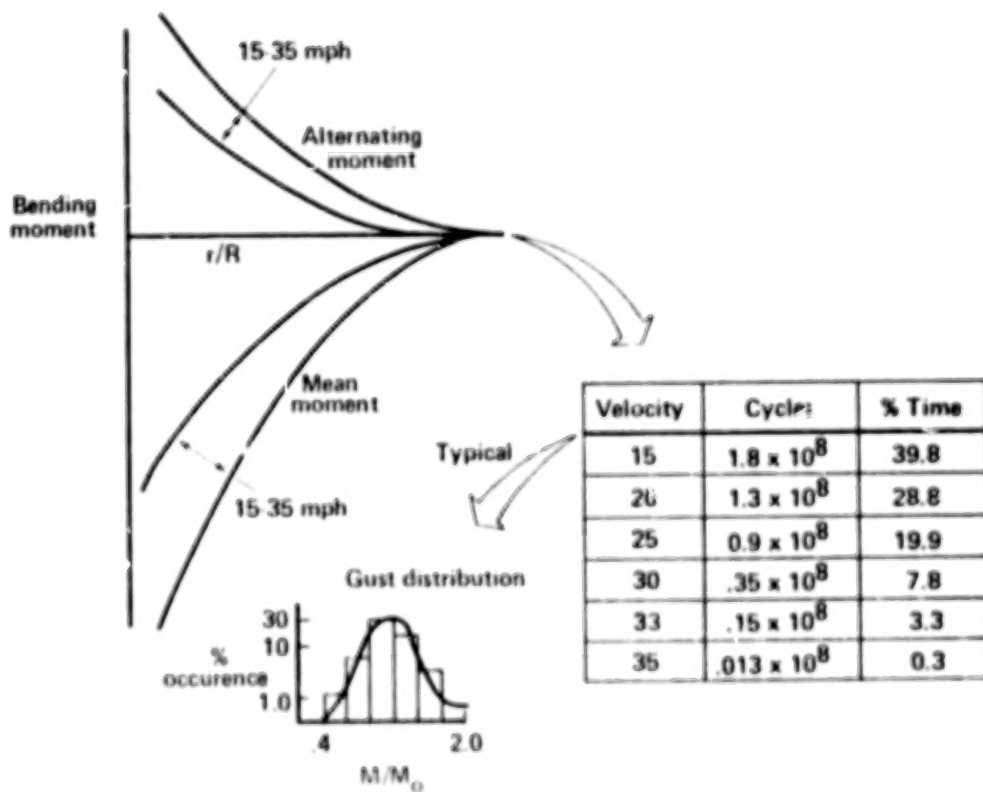


Figure 11. Fatigue Load Spectrum

Type of welded structure	Weld category	Crack growth design allowable flaw size		Inspection method & flaw detection capability		Flaw size acceptable criteria
		Surface	Internal			
Upper surface Chordwise welds	B (16,000 psi allowable)	.06 deep X .30 long	.12 deep X .30 long	VT	.005 wide X .06 long	.06 long (linear indications) .125 long (round indications)
Transition section welds				PT	.005 wide X .03 long	
Trailing edge spanwise Welds sta 51 to sta 250				RT	2% of t deep X .04 long	.125 long
				UT	.03 deep X .09 long	
Lower surface Chordwise welds	C (12,000 psi allowable)	.09 deep X .44 long	.18 deep X .44 long	VT	.005 wide X .06 long	.06 long (linear indications)
T-stiffener Spanwise welds				PT	.005 wide X .03 long	
Trailing edge Spanwise welds	E (5,000 psi allowable)	.54 deep X 2.70 long	1.08 deep X 2.70 long	VT	.005 wide X .09 long	.125 long
Leading edge Outboard of sta 250				PT	.005 wide X .06 long	

Figure 12. MOD-1 Spar Welds Inspection Matrix

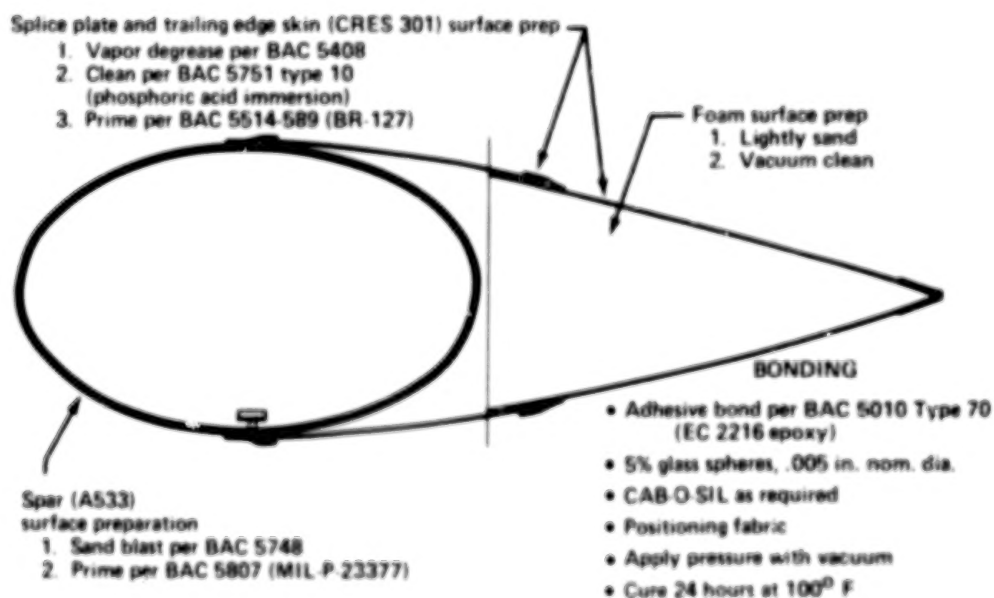


Figure 13. Surface Preparation and Bonding

Test	Environmental conditioning	Test temperature	No. of specimens	Results
Tensile (lap shear)	120°F, 100% RH, 72 hrs	Room	5	3,750 psi avg (cohesive failure)
	None	-31°F	5	2,992 psi avg (adhesive failures)
	None	Room	5	3,050 psi avg (cohesive failures)
	None	125°F	5	1,720 psi avg (cohesive failures)
Creep tests	None	5 hrs at 200°F	4 cycles (from 125°F)	No elongation at 2.4 psi
		5 hrs at 230°F	3 cycles (from 125°F)	No elongation at 1.1 psi

Notes: Selective tests with Cab-o-Sil added for viscosity control showed no strength degradation.  
Tensile tests after creep cycles averaged 4,040 psi.

Figure 14. Bond System Verification

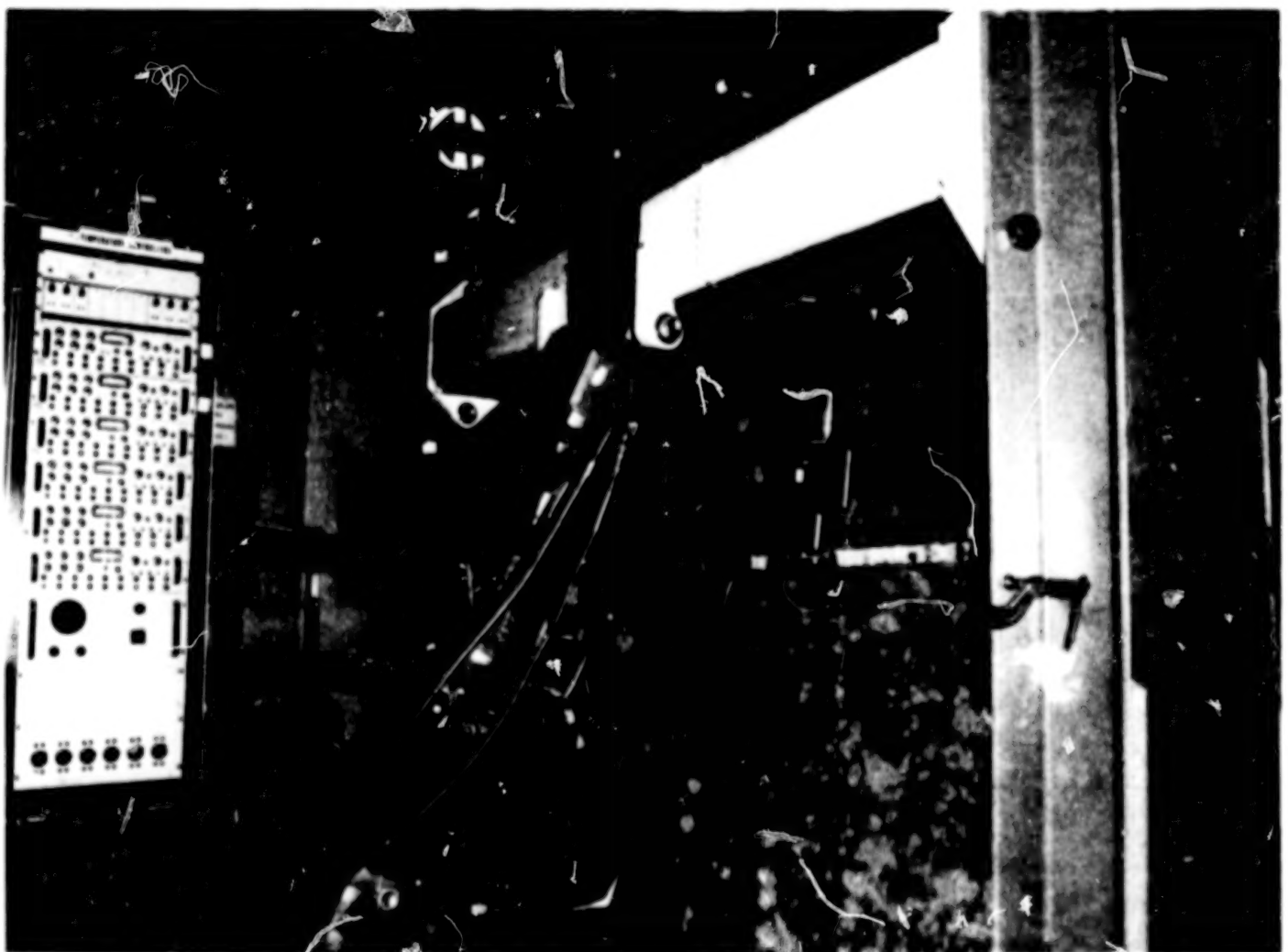


Figure 15. TE Fatigue Test Setup.

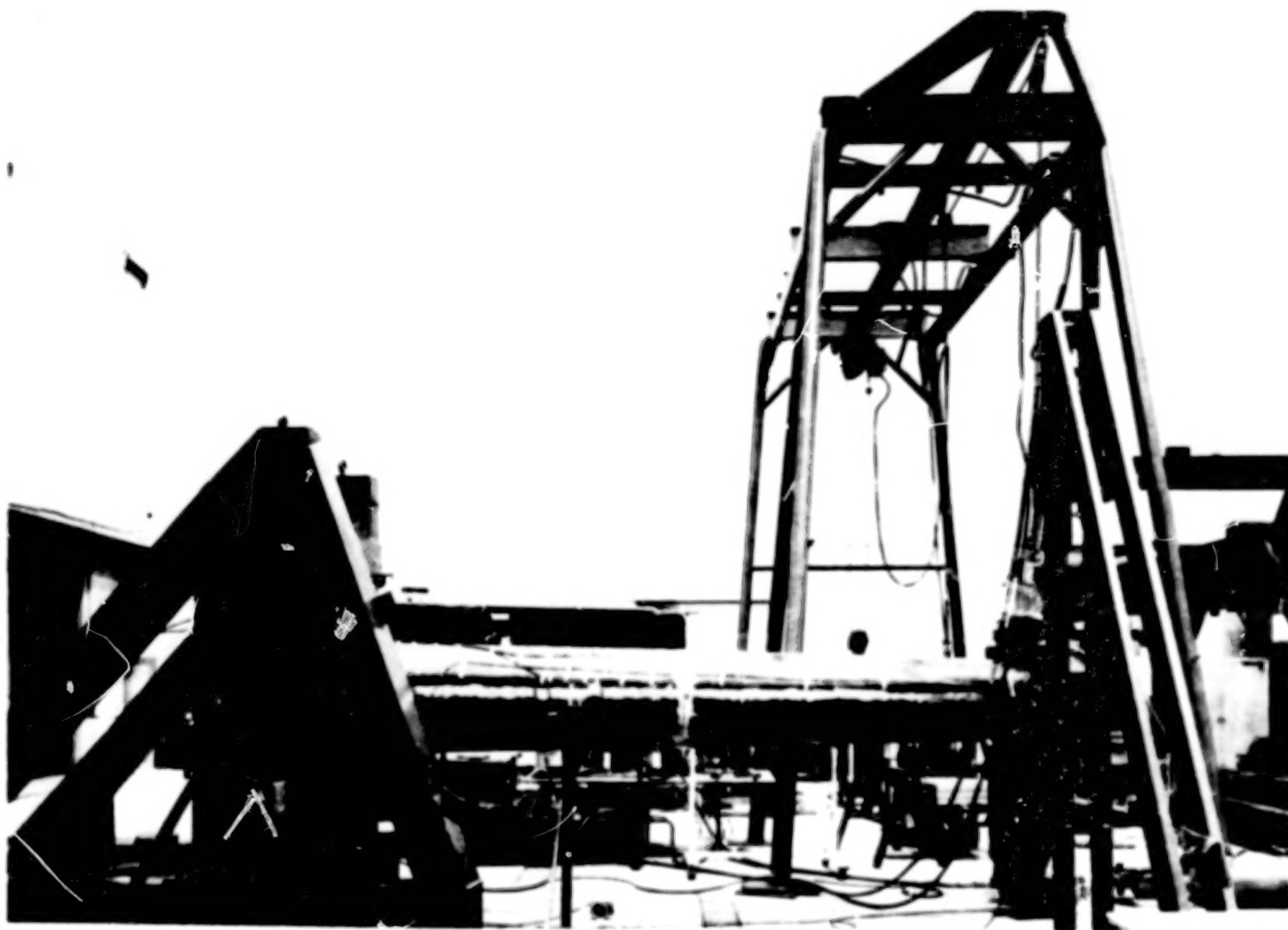


Figure 16. Spar Section Bending Test.

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- SPAR (17,000 lbs)  $F_{TY}=70$  ksi
  - High strength steel
  - Weldability and formability
  - Notch toughness
  - Base metal fatigue allowables  $S_R=28.6$  ksi
    - High quality steel
    - Controlled surface finishes
    - Tapered tension side skins
    - No mechanical fasteners (no holes)
  
- SPAR (17,000 lbs)
  - Weld area fatigue allowables  $S_{R(B)}=16$  ksi
  - Weld detail per AISC spec  $S_{R(C)}=12$  ksi
  - Post weld heat treatment  $S_{R(E)}=5$  ksi
  - Sculptured tension skins
  - Nuclear quality weld
  - Multiple NDT of welds (UT, PT, RT, VT)
  - Buckling allowables  $F_B=56.9$  ksi
    - Column stiffener (test result)
  
- TRAILING EDGE (3,000 lbs)  $F_{TY}=90$  ksi
  - Stainless steel skins— $\frac{1}{4}$  H-301
  - Induced stresses + airloads
  - Density optimized foam core
  - Modulus to support skins (face wrinkling)
  - Lightening holes

Figure 18. MOD-1 Blade Design Solutions.

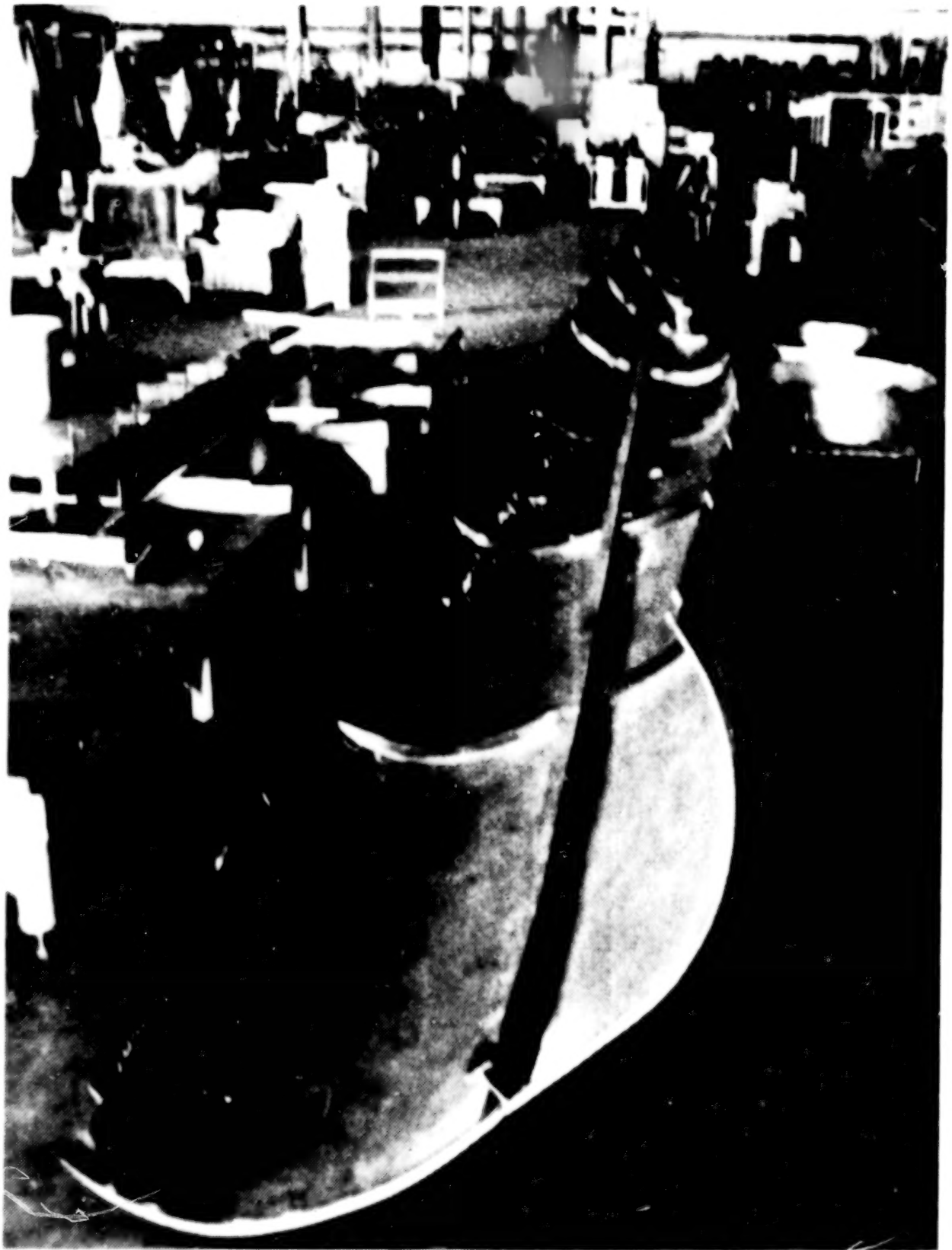


Figure 19. Twisted Weldment.

## THE BOEING MOD-2 WIND TURBINE SYSTEM ROTOR

Gordon N. Davison  
Boeing Engineering and Construction Company  
Seattle, Washington 98124

### INTRODUCTION

In a preceding discussion, Mr. Richard Douglas described the several very important trade studies which led to the final design stage of the MOD-2 Rotor. The following discussion includes the design details, significance of fatigue strength, design development test results, and conclusions of the preliminary design efforts.

### SPECIFICATIONS AND REQUIREMENTS

The outstanding configuration requirement for the MOD-2 Rotor is the 300 foot diameter listed in Figure 1. It may well be the practical size limit of the future wind turbines even though no specific restraints were encountered in its design development. Rotor bending frequencies do, however, tend to reduce with increasing span and the Rotor, drive train, and tower frequencies, become more difficult to separate from the two and four per rev forcing frequencies.

A second noteworthy feature of the MOD-2 is the Rotor's controllable tip. The 30% semi-span tip for each blade involves a bending moment which must be carried through the spindle. This bending moment is large enough to present interesting considerations in the detail design of the spindle assembly and its support.

The design load conditions are derived from both the environmental and functional requirements and may be categorized as being normal operating, operating fault, and non-operative. Both the limit design (static load) and the cyclic conditions of normal operation involve essentially the same environmental factors, i.e., steady winds, wind gradients, and gusts. The normal operation therefore governs the approach to fatigue restraint design while the non-operative conditions, such as the extreme winds, are the critical aspects of the buckling resistant design criteria.

Cost penalties associated with a minimum weight rotor were avoided by using the "soft" tower concept. Although the rotor speed passes through the rotor/nacelle/tower combined natural frequency during startup and shutdown, the dwell times are short and not detrimental. Weight restraints on the rotor are less severe than if the tower were "stiff" designed.

In order to assure the adequate fatigue strength for the 30 year lifetime of the Rotor, the number of joints designed into the skin panels is being held to a minimum. The number of joints is minimized by the use of maximum material sizes, consistent of course with forming capabilities. For general cost reduction, there is a requirement inherent in the MOD-2 Rotor for design simplicity, an inexpensive, easily welded steel alloy, and a minimum number of parts.

#### DESIGN

The final MOD-2 Rotor configuration, MOD-2-107, is composed of three sections as shown in Figure 2. The view is taken from the nacelle side and shows the down-going portion of the Rotor. The tip section is 45 feet, the mid section is 75 feet and the hub section is 60 feet long. The tip chord is 56.6 inches and the maximum chord, throughout the hub, is 136 inches. The maximum airfoil thickness varies from 6.78 inches at the tip to 57 inches at the hub.

When the controllable tip section is in the high drag, or feathered, position the tip section trailing edge is down-wind as shown in the small view labeled "High Wind Attitude". Station 360 is made into a bolted field joint to permit interchangeability of blade elements in case of damage and to facilitate shipping. The hole in the near side of the Rotor at Station 0 is the accommodation for the teeter assembly and stub of the low speed shaft.

The spindle installation is shown in Figure 3. The angle of attack is controlled by means of the tip actuator pushing against the stub fitting on the trailing edge of the tip. The spindle assembly is pressed into rib fittings in the tip and bolted to the inboard rib of the tip. The spindle rotates on two lubricated roller bearings located in the outboard end of the mid section. Access to the bearings for maintenance and to the control system is gained through small doors and hatches in the low stress areas of the compression skin panel. A non-structural fairing covers the entire cavity between the tip and mid sections.

The mid section is characterized by a very long, simple, formed skin construction terminating in a machined rib at Station 360. Considerable analysis has gone into the design of the back-to-back field joint ribs at Station 360 because of the obvious fatigue-critical nature of such a joint. The thickness of the joint flange is 1.25 inches.

The hub section completes the transition from the air foil shape of the blade to the non-lifting contour shown in Figure 4 in the central region of the hub. The hub section contains the teeter assembly which is crucial to the relief of high frequency cyclic loading to the low speed shaft. The Rotor is actually supported by two elastomeric bearings, which in turn transmit the dead weight, shear, bending and torsion into a teeter trunnion. The trunnion is welded to the stub of the low speed shaft. A teeter stop limits the travel to  $\pm 5$  degrees, and an associated teeter brake will hold the Rotor and prevent flapping motions when the Rotor is parked.

The field joint on the low speed shaft in Figure 4 is used for final assembly when the Rotor is hoisted horizontally 200 feet up to the nacelle. All of the electrical, hydraulic, and pneumatic subsystems enter the Rotor from the inside of this low speed shaft.

#### DESIGN PROCEDURE

The evolution of the MOD-2 Rotor to the status in preliminary design used load and material thickness iterations varying from rather simple analyses to sophisticated structural dynamics computer programs. For the Rotor geometry selected as a result of the trade studies early in the program, limit wind conditions corresponding to a hurricane acting uniformly across the Rotor disk and 9999% cumulative probability of occurrence of gusts during operation were used to generate a bending moment distribution. The plate thicknesses, based on certain plate lengths (joint locations), were calculated from static buckling allowables and, with the resulting stiffness distribution, new iterations of loads, thicknesses, joints and allowables were performed. Since fatigue considerations were paramount, the following discussion of fatigue load spectra, fatigue allowables based on a fracture mechanics approach, and pre-flawed specimen testing summarizes the procedure to obtain a conservative fatigue resistant design.

Cyclic blade loads are caused by rotation in a gravity field, wind gradient with yawed flow, by wind gusts, and by startups and shutdowns. The magnitudes of these cyclic loads are dependent on Rotor characteristics, such as weight, hub restraints (teetering), lift and drag forces, and natural frequencies. They are also dependent on wind characteristics, such as steady operating wind speeds and associated turbulence. For the MOD-2,

all wind turbulence up to and including gusts having a 99.9 percent cumulative probability of occurrence are included in the design environment. The number of cycles of loading are dependent on the design life of the MOD-2 (30 years), the rotating speed (17.5 RPM), the probabilities of occurrence of various wind conditions, and the ability of the MOD-2 design to attenuate to insignificant levels the response to low and high frequency turbulence. The attenuation is accomplished with an active control system for low frequency gusts, and the teetered hub for high frequency gusts. The resulting fatigue stress spectrum for one point on the blade is represented schematically in Figure 5. Three types of stress cycles are defined. Type I cycles are due to rotation in wind gusts at a given steady wind speed. Type II cycles are steady stress transition cycles due to gusts. Type III cycles are due to WTS startups and shutdowns.

The allowable fatigue stresses are calculated using the fracture mechanics pre-existing flaw approach. This assumes a crack-like defect exists in each critical area of the structure from the first day of operation. The behavior of the assumed pre-existing crack is characterized by a crack growth model which predicts the growth of the crack from initial size to failure. The crack growth model utilizes the stress intensity concept in predicting the crack growth behavior. The relationship between the characteristic initial stress intensity and the number of cycles to failure is shown in Figure 6. The initial flaw assumed in the analysis is larger by a factor of 2 or 3 than the minimum size which can be reliably detected during normal inspections. The conservatism in the initial flaw size assumption, therefore, translates to a significant margin in both allowable stress and life.

Verification of the crack growth model used in the determination of allowable stress levels was accomplished by spectrum load tests of pre-flawed (0.25 X 0.05 inches) specimens. A total of four different spectra were tested in order to provide a data base which would encompass variations in design load spectra.

Correlation between predicted and actual results is presented in Figure 7 where each bar represents a test point. The data points are evenly disbursed about 1.00, with an average correlation of 1.00, which means the actual and predicted stress are identical. The majority of the tests were conducted on other ASTM A Type steels. The excellent correlation between predicted and actual results verifies the applicability of the crack growth model for wind turbine type load spectra and A Type steels.



Although the same number of discrete load cycles are applied at each point along the Rotor, the magnitudes of the steady and alternating stresses vary from point to point. Since the allowable stresses are determined by establishing the crack growth behavior of an assumed initial flaw, the allowable stress changes as the imposed stress spectrum changes. There is a significant difference in stress spectrum at the hub relative to that at the tip of the blade. The allowable stress level by Rotor Station presented in Figure 8 reflects the changes in stress spectrum. The step decrease in allowable at Station 90 is an added conservatism to account for the redistribution of loads because of the change in sections at that station.

With respect to the maximum static load conditions, the Rotor is critical in buckling at several locations. The combination of curved and flat plates in a bending situation, such as exists in the MOD-2 blade preliminary design, presents an uncertainty in determining the end fixity of the curved panel and therefore the determination of the buckling strength of the section. The analysis for the MOD-2 blade design assumed a conservative approach to determine the buckling allowables. A blade bending test was desired to verify the analysis methods used for buckling under axial and bending compressive loading.

A full scale section representing the MOD-2 blade at preliminary design was prepared using the forming and welding procedures developed for fabricating the prototype Rotor. This section, 35 feet in length, represented the geometry from the Station 360 flanged field joint to Station 780, as shown in Figure 9. In the interest of economy, selected plates were standard thicknesses and therefore deviated slightly from preliminary design sizes. This had no effect on the objective of the buckling program.

An applied load transfer rib was added at one end and a skin thickness increase was built into the transition region at the root of the specimen. The test specimen was fabricated by the MOD-2 manufacturing shop and was available for the test on November 3, 1978. Test fixtures were designed and fabricated which would permit interfacing the specimen with the structural test strongback. Instrumentation of the specimen, installation on the strongback, and the completion of the test set-up is shown in Figure 10. The test consisted of applying a panel compression in the form of a couple and a shear load at the outboard end on the specimen. These loads were applied simultaneously and as percentages of the test load.

The specimen was subjected to 148% of the predicted ultimate strength without evidence of buckling. The primary objective

the test was accomplished in that the buckling analysis method was shown to be conservative. The preliminary design of the blade was shown to have comfortable margins from the standpoint of buckling.

Another important aspect of the development program was the actual manufacturing experience. Considerable confidence and knowledge was gained in the forming, welding, and handling of the blade elements. Since the blade section was not tested to destruction, it was also possible to perform a post-weld stress relief heating cycle. Measurements made before and after the heating cycle showed no change in the specimen shape.

#### CRITICAL FACTORS

The concept of forming the skin panels of an all-steel, all-welded Rotor into an airfoil shape on large brake presses leads to great simplification in the overall manufacturing process. The feasibility of such operations was established on the MOD-1 blade. The technique has been improved on the development section for the MOD-2 Rotor.

The fatigue resistance of the MOD-2 Rotor joints is also of considerable importance. In addition to flaw growth development testing, and extensive fatigue load spectra analysis, a manufacturing plan has been developed which will permit fabrication of the critical tension skin joints in tooling with easy access for inspection and repair. At other places in the Rotor where ultrasonic and radiographic inspection are not feasible, Class C weld allowable stresses have been used in sizing structure.

#### COST DRIVERS

There are several factors that figure prominently in the cost of fabricating the MOD-2 Rotor. They are, not necessarily in order of importance; tooling, weld quality, weld length, and machined parts. There have been concerted efforts to reduce the cost of each of these in relation to the others, since all are interrelated.

The significant efforts on each were:

1. Tooling

Adequate tooling to provide the necessary support for skin panel assembly and hard points at the ends of each section while minimizing final assembly tooling.



2. Weld Quality  
Reduction in the number of weld joints requiring ultrasonic and/or radiographic inspection, and maximizing the number of fillet versus groove welds.
3. Weld Length  
Finding suppliers who could furnish large plates to minimize the number of panel joints as well as finding the largest brake presses available for forming wide panels.
4. Machined Parts  
Minimizing the number of machined parts by use of welded assemblies and by simplifying the shape of the machined ribs to allow the use of two-axis milling machines.

#### MAJOR PROBLEMS

There are no major problems at the present time and none are foreseen. There is a need to proceed to prove the adequacy of design and cost analysis by actual fabrication and operational experience.

#### CONCLUSIONS AND RECOMMENDATIONS

It is concluded there are no specific restraints relative to maximum Rotor diameter. A controllable tip is the most significant complexity in the design of a large diameter Rotor. The tension skin joints are the most critical weldments in the Rotor and will be inspected for conservatively small flaws. The compression skin panels have a conservative buckling allowances as demonstrated by development test.

The bending frequency of the tower has been deliberately designed below that of the Rotor. This separation of frequencies is adequate enough to permit stiffness and weight changes in the Rotor which do not cause a deleterious coupling of the Rotor and tower at the two per rev and four per rev forcing functions.

It is strongly recommended that rotor, drive train, and tower designs proceed concurrently. Design decisions for each, which interact with other systems, may be made at least-cost and least-impact on any one component.

- External configuration requirements
  - Rotor diameter  $\geq 300$  ft
  - Airfoil contour = NACA 230XX
  - Twist about 50% chord =  $-2.5^\circ$  to  $+4^\circ$
  - Controllable tip = 30% semi-span
  - Pitch control =  $+5^\circ$  to  $-97^\circ$
  - Teeter =  $\pm 5^\circ$
- Environmental requirements
  - Design rotational speed = 17.5 rpm
  - Cut-off wind speed @ hub = 45 mph
  - Steady winds plus gusts
  - Lightning, temperature, precipitation, projectile
  - Non-operative: snow, ice, extreme winds
  - Handling and transportation
- Internal design requirements
  - Weldable, low cost steel construction
  - Commercial tolerances
  - Limit operating loads
  - Fatigue loads, 30 year life
  - Operating fault loads: overspeed, inadvertant feathering and braking

Figure 1. Specifications and Requirements

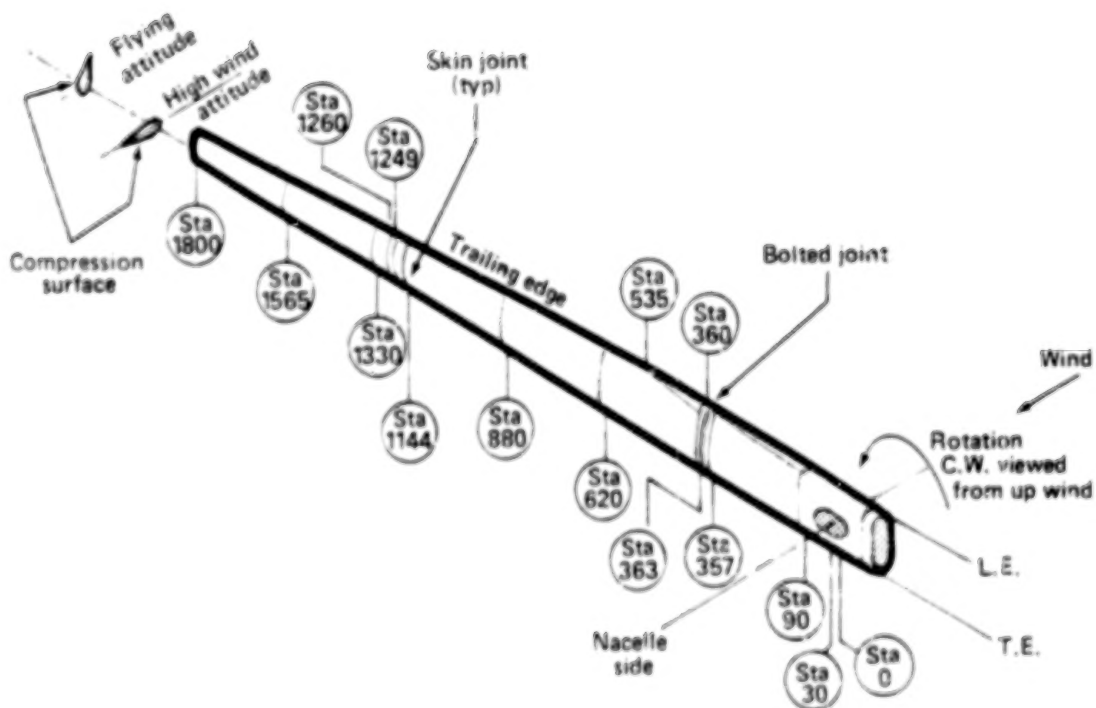


Figure 2. Rotor Configuration MOD-2-107

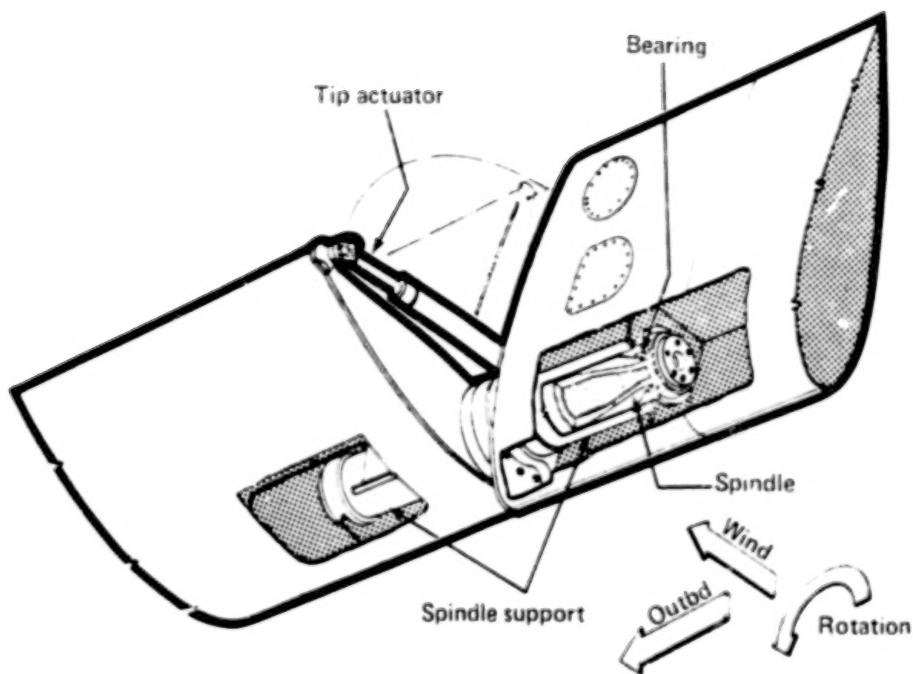


Figure 3. The Spindle

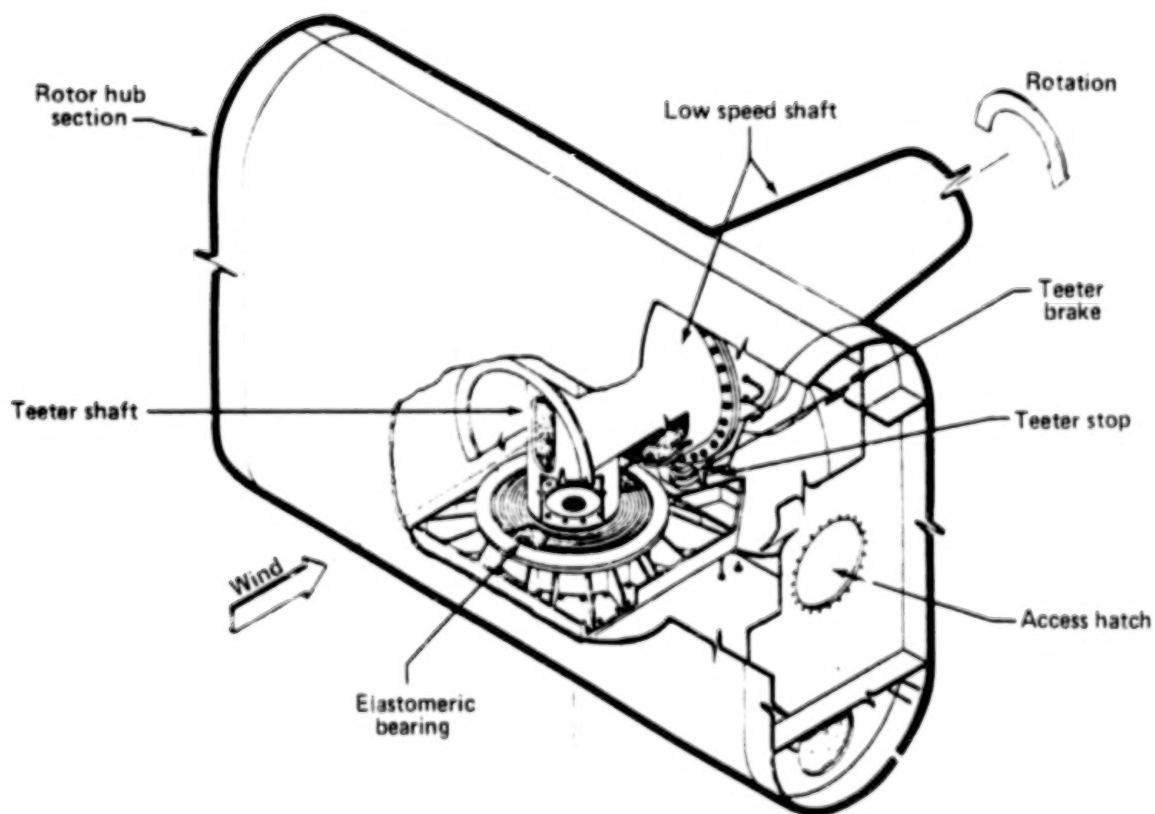


Figure 4. The Hub

For each point on the rotor:

- Type I - one per rev alternating stress during gusty winds
- Type II - steady stress transitions due to gusts
- Type III - startup and shutdown stress cycles

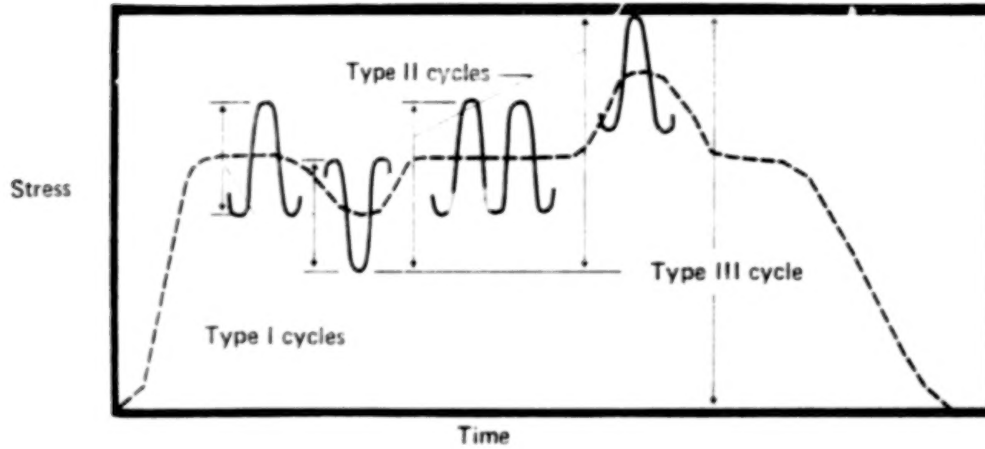


Figure 5. Rotor Blade Fatigue Spectra

- Model derived from spectrum load test results
- Model is good for all A type steels
- Model applicable to all wind turbine spectra

Crack growth model

- Accounts for threshold effects
- Accounts for retardation effects
- Predicts constant amplitude data

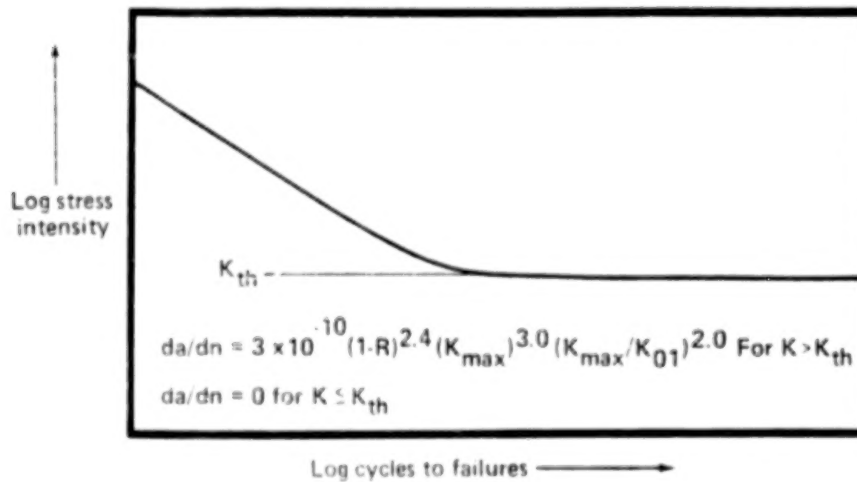


Figure 6. Fatigue Allowable Model

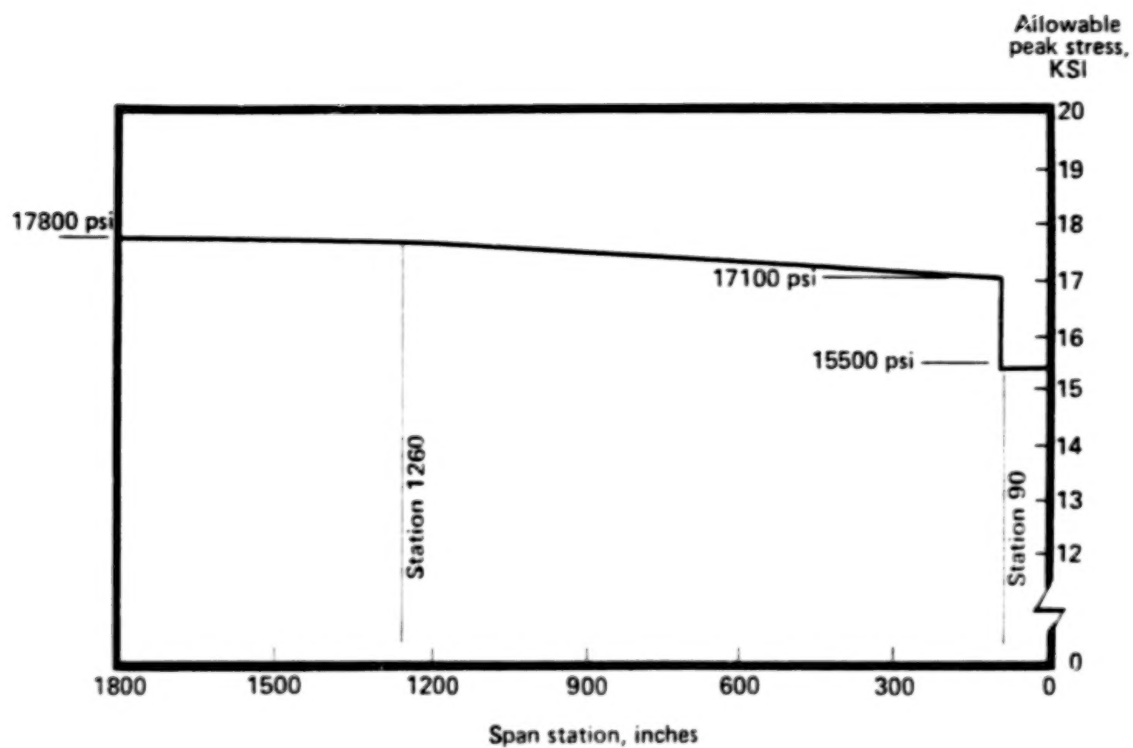


Figure 7. Rotor Fatigue Allowable Stresses

- Each bar represents a test data point
- Except as noted the test material was A533

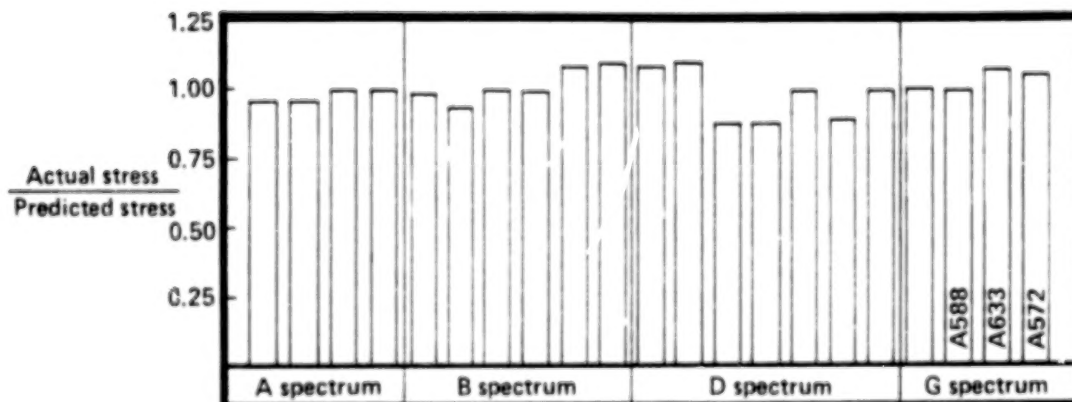
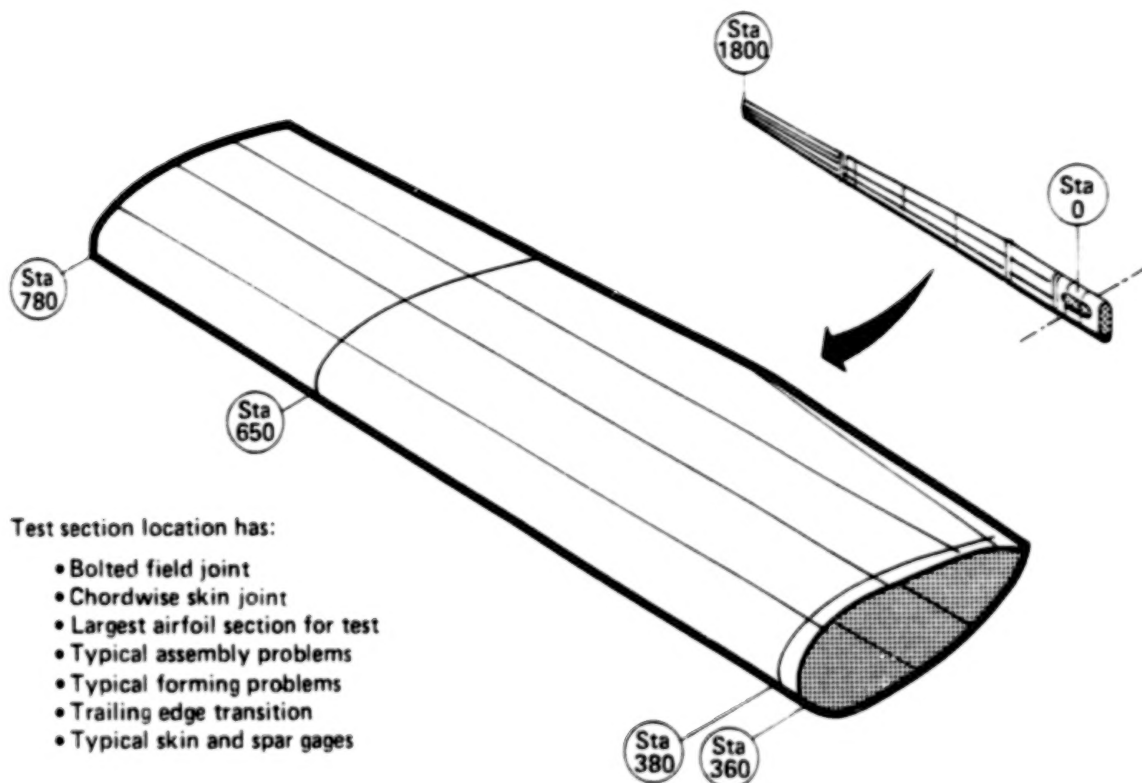
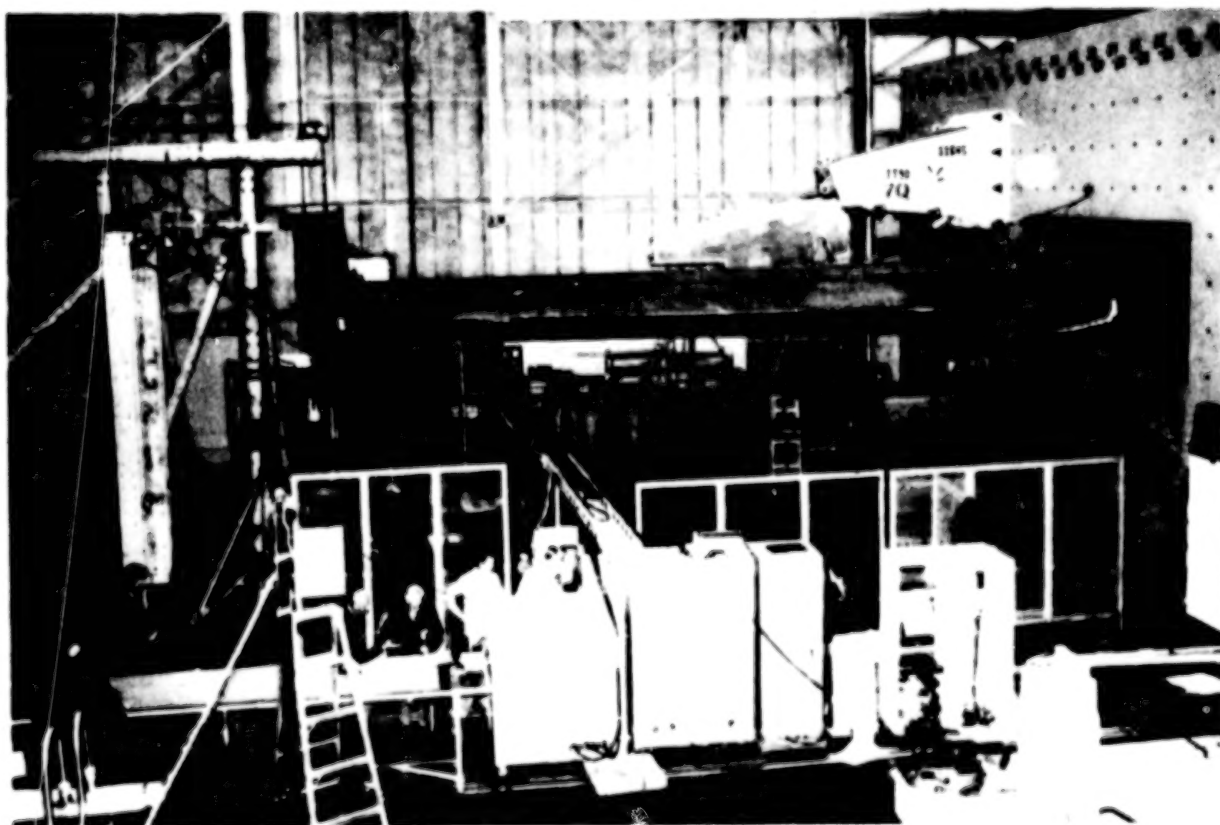


Figure 8. Correlation of Test and Predicted Results



*Figure 9. Rotor Blade Development Bending Test Specimen*



*Figure 10. Rotor Blade Development Bending Test*

## STATUS OF THE SOUTHERN CALIFORNIA EDISON COMPANY 3 MW WIND

### TURBINE GENERATOR (WTG) DEMONSTRATION PROJECT

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To demonstrate the concept of utility scale electricity production from a high wind energy resource, Southern California Edison Company (SCE) has initiated a program to construct and test a 3 megawatt (3,000 kW) Schachle Wind Turbine Generator (WTG) at a SCE-owned site near Palm Springs, California. The background and current status of this program are presented along with a summary of future planned program activities.

\* \* \* \* \*

With fossil fuels becoming increasingly expensive and in short supply, the wind is showing increasing promise of making an important, cost competitive contribution toward meeting the public's needs for electrical energy. Southern California Edison (SCE) has become especially interested in wind energy conversion for two primary reasons. First, over two-thirds of SCE's generating capacity consists of oil-fired units, and consequently SCE has a very high reliance on imported low sulfur fuel oil with the attendant problems of high fuels costs and embargo possibilities. Use of WTG's could help reduce this dependence on oil. Secondly, wind speed data obtained from a Department of Energy (DOE) meteorological tower show that there is an outstanding wind resource area near existing SCE substation and transmission facilities in the vicinity of Palm Springs, California (fig. 1).

These factors led SCE to undertake evaluations of WTG's (including designs of both DOE and private suppliers) for possible use on the SCE system (fig. 2). One of the private designs evaluated was that of Charles Schachle, Wind Power Products Company. The Schachle design had been under development since 1970, well before the beginning of the Federal Wind Program. Over three years were spent on blade design alone. These efforts culminated in construction of a prototype system with a 72 foot blade span which has been operational since May 1977 in Moses Lake, Washington (figs. 3 and 4). Results of SCE's evaluations of the Schachle WTG design indicated that it was an especially promising design for relatively near-term economical electricity production. Consequently, SCE decided to purchase and test a 3 mW Schachle WTG (fig. 5) at the SCE-owned test site near Palm Springs (fig. 6).

The purpose of the project is to demonstrate the technical and economic feasibility of the Schachle WTG as a necessary intermediate

step toward the more widespread installation of such WTG's to generate electrical energy for the SCE system on a commercial basis. It will also serve to give the public a first-hand opportunity to visualize how wind-generated electricity may contribute to their future energy needs.

Unique features of the Schachle WTG include a tubular conically shaped steel tower mounted on a concrete and steel base which will permit tower rotation to keep the blades facing into the wind. The blades will be constructed of laminated wood and fiberglass. The WTG also uses a hydraulic pump-motor link between the rotor (atop the tower) and the generator (at ground level) which allows a less complex gearbox design and permits the rotor blades to rotate more efficiently at speeds varying in proportion to the speed at which the wind blows. The unit will have a 165 foot blade span and a design rating of 3 megawatts in a 40 mph wind.

The unit was rated at 40 mph to maximize the energy (kW hr) output for the specific wind regional experienced at the SCE wind site near Palm Springs. Based on a significant quantity of wind data, it was determined that a unit rated for peak output at 40 mph would produce twice the energy of one rated at 25 mph at the site being considered. The value of the increased energy more than offsets the increased cost of the higher rated WTG. The unit will operate unattended and is expected to produce 6,000,000 kW hr/year in the winds present at the test site. This is enough electrical energy to supply the average annual needs of 800-1000 customers, and would save about 10,000 barrels of oil a year which would otherwise be used to generate this amount of electricity.

The total cost of the project is about \$2,000,000, of which \$1,000,000 is for the WTG unit and \$1,000,000 is for SCE's site work, substation and other interface equipment, test equipment, design review, and related expenses. The project is financed solely by Southern California Edison.

After obtaining the necessary siting permits, site preparation was begun in early December 1978, and was completed in early January 1979. Site fencing was complete by the first of February. The substation at the site was completed in late March.

The installation of the WTG foundation is projected to be complete by May. The tower and blade assemblies will then be installed at the site. The complete system including WTG, substation, control system, transmission line tie-in and test equipment is expected to be operational around mid-1979.

At that time a one- to two-year performance test program will be initiated. The purpose of the test program is to obtain operating data to evaluate various performance characteristics of the WTG and to determine its operating and maintenance characteristics. Data obtained will be recorded on magnetic tape for subsequent processing and analysis.



Data to be recorded include such items as wind speeds and corresponding mechanical and electrical output, blade and tower loadings, operation of the control system and other related parameters. Maintenance and overall reliability data will be taken. Data will be taken to determine whether there are any significant levels of noise, television interference, or impact on birds or other biota associated with operation of the WTG.

The WTG and data recording system will be capable of unattended operation, automatically coming on-line whenever adequate winds, above 10 to 12 mph, are present. Southern California Edison personnel at Devers Substation, a major substation 1/4 mile from the site and manned 24 hours a day, will monitor the operation of the WTG.

The data from the test operation of the WTG will be analyzed from technical, economic, and environmental standpoints. Results of the analysis of the WTG performance testing will be used in ongoing studies to evaluate the potential for installation of large arrays of such WTG's for commercial production of electrical energy in the high wind areas near Palm Springs. These studies will also incorporate the results of a current wind energy resource evaluation project jointly sponsored by Southern California Edison and the California Energy Commission in which wind data is being continuously monitored at 19 sites in the area.

As other promising WTG's, such as the DOE MOD-2 or other DOE or private designs, become available, SCE would be seriously interested in finding ways to use its wind test site for comparative testing of such designs in addition to the Schachle WTG.

The purpose of such testing is to develop as early as possible proven competitive designs which will serve to bring the price of wind turbine generators down to the lowest possible value. If the results of Edison's WTG tests and wind data collection efforts verify expectations, the first commercial implementation of large-scale WTG's could be a reality on the SCE system in the 1980's. The initial units would likely be installed in the high-wind areas in north central Riverside County near Palm Springs, but other areas will be explored which may also have the high winds necessary for efficient WTG operation, and later units could be installed in those areas.

There are a number of practical aspects that must be kept in mind when considering the implementation of WTG's for commercial energy production. First of all, even very large WTG's can produce only a few megawatts of electrical power each, and large numbers of them (800-1300) would be necessary to match the energy output of a single typical conventional fossil or nuclear fueled generating station. In this regard, it is exceedingly important to demonstrate that large arrays of WTG's can be installed in a way which is acceptable to the public. Secondly, the output of a WTG is very much dependent on the winds available at the site, and great care must be taken to select a site with an adequate

wind resource. Remote sites requiring long transmission lines to transport the energy to the customers may not be economically viable even if the resource is good. Thirdly, because the wind is not always available upon demand, conventional generation sources will continue to be required to provide back-up capacity so that customer electricity requirements are reliably met.

If progress in the general field of wind energy conversion continues, and if the performance of the SCE 3 WTG meets expectations, wind energy has the promise of providing a valuable and significant contribution to the energy needs of SCE's customers possibly as early as the 1980's. However, SCE does not consider wind to be the complete answer in itself. SCE must employ wind and other new alternate energy sources, along with conventional sources in appropriate combinations, to adequately meet its customers' needs for reliable and affordable energy for their homes and jobs.

#### Discussion

- Q. What kind of a bearing plan or track plan is designed for the base of the tower rotation? It looked quite weak.
- A. It is very similar to that used on Schschle's prototype, as shown in previous slides. The track will be upgraded to match the strength that is required for the 3 MW unit.
- Q. What is the overall efficiency of the hydraulic pumping and motor cycle? Is there a value for the prototype?
- A. We don't know the exact number, but we are looking at possibly a 25 percent loss in the hydraulic portion of the system. That matter will be evaluated in very strict detail during the operational testing program. The prototype utilized some off-the-shelf equipment not optimized for the WTG. I think the losses there are somewhat greater than the 25% value expected for the 3 MW unit.
- Q. The nacelle and rotor seem to be mounted forward pretty much into the wind. Is there a tendency to tip if the wind should have a 180 degree reversal? Is there a provision in the rear mount to provide negative load?
- A. Even with the wind from the rear, there should normally be no up forces on the rear leg. If in an extreme case up forces are encountered, they will be accommodated by the center pivot. The unit is designed to withstand rear winds comfortably in excess of 100 mph.

Q. Did you consider a cost tradeoff for not putting the yaw on the tower?

A. That's something we have been considering, but we have not addressed the concept in detail. In the Palm Springs area where our site is located, most of the strongest winds come from the west, and therefore, it might be cost-effective to design a simplified machine without yaw capability. That is a good point, and we will be investigating it in more detail in the future.

Q. There seems to be a consensus that a two-blade machine is cheaper than a three-blade machine. Could you elaborate why you decided on a three-blade machine?

A. That is a design decision made by Mr. Schachle. I believe the primary reason for three blades was to reduce dynamic loads, and Mr. Schachle chose three blades as the overall most cost effective approach for his specific wind turbine generator design.

Month	Miles per Hour
J	5.5
F	10.5
M	18.5
A	17.5
M	22.5
J	19.5
J	19.0
A	15.5
S	17.0
O	10.5
N	10.0
D	7.5

## SCE PLANNING



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SCHACKLE WIND TURBINE GENERATOR (Moses Lake, Washington)



Figure 3



Figure 4



**FUTURE  
SCE  
WIND  
ENERGY  
TEST  
UNIT**

Figure 5

SAN GORGONIO PASS (looking west)



Figure 6

## RESULTS OF A UTILITY SURVEY OF THE STATUS OF LARGE WIND TURBINE DEVELOPMENT

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S. Quraeshi, Shawinigan Engineering Co. Ltd.  
Montreal  
L. P. Rowley, Canadair Ltd., Montreal

### INTRODUCTION

A survey of the status of wind turbine development has been commissioned by a utility company having interest in the application of wind power to the generation of electricity. The utility, Hydro-Quebec, is considered one of the major utilities in North America in that it services one of the largest geographical land masses in the continent. An indication of the scope of the utility is given by Table I.

### SURVEY TEAM

The survey was carried out by a group of researchers drawn from three distinct and complementary fields as shown below.

<u>Type</u>	<u>Organization</u>	<u>Related Work</u>
Consulting Engineering	The Shawinigan Engineering Co. Ltd., Montreal.	Energy and Power Consultant, Design, Engineering and Project Management of Power Projects for Utilities.
Utility	L'Institut de Recherche de l'Hydro-Québec, Montreal. (IREQ)	R & D on energy production, transmission, storage and utilization.
Manufacturing Industry	Canadair Limited, Montreal.	Analysis, Design and Manufacture of aerospace structures.

### Related Team Experience

- . IREQ work on wind energy since 1970
- . Design and operating experience of 230 kw VAWT
- . Design studies on units rated up to 10,000 kw

## METHODOLOGY

The overall objective was established as:

Examine Wind Energy Conversion Systems (WECS) from a utility viewpoint to establish the state-of-the-art with regard to:

- . Availability of the type of machines
- . Quality of power generation
- . Suitability of WECS for electrical grid systems
- . Reliability of the WECS
- . Economics

The methodology employed consisted of:

- . Survey of literature in the public domain
- . Visit to installations of interest in Europe and North America.
- . Discussion with Designers, Operators and Planners
- . A design study involving state-of-the-art WECS

## RESULTS

Table II provides summary data for those wind turbine generators (WTG's) covered by the survey and includes machines which exist or are projected, and for which data is available. Those machines marked \* were visited by representatives of the three companies. The personnel concerned with demonstrating and discussing the various machines were most co-operative and their assistance is gratefully acknowledged. Figure 1 presents a size comparison of six of the large (over 1000 kw rating) HAWT's in the survey.

Figure 2 presents a size comparison of eight smaller HAWT's.

Figure 3 shows a size comparison of the VAWT's included in the survey. Note the scale of the smaller machines.

Figures 4 and 5 show schemes for large VAWT's conceived as proposals for the Canadian Government.

In addition to the HAWT's and VAWT's the study briefly examined four innovative concepts, the Vortex Augmentor, the Tornado Tower, the Madaras Rotor and the Diffuser Augmented Wind Turbine.



## DISCUSSION OF RESULTS

Of the twenty three designs listed in Table II, seven are VAWT's, nine are upwind HAWT's and seven are downwind HAWT's.

Of the total group, seven are of a rating which would be of real interest to a utility (Figures 2 & 4) and, of these, one is a VAWT, (the Canadian Government Proposal); four are upwind HAWT's and two are downwind HAWT's. The largest of the group, the German GROWIAN is a design concept, while the Wind Power Group unit is in the design stage. Three machines, Mod 1, Mod 2, and the WPPC design are in the manufacturing/installation stage. Only one, the Danish Tvind HAWT is actually in operation. As of October 1978, this unit had been operated up to 400 kw, 23% of its rated power, and so far as is known is the largest unit both in size and design rated power in existence.

The smaller HAWT's (below 630 kw), shown by Figure 3, range from the Danish Gedser machine with its derivatives Mod A and Mod B, through the Mod 0, Mod 0A series and the WTG Cuttyhunk machine, to the FDO unit being designed in the Netherlands. The lower end of this scale is occupied by the Saab-Scania unit which is a test bed for the 2000-4000 kw Swedish Government design study, and the Lawson-Tancred machine in the UK which has an innovative transmission system and power storage facility.

VAWT's are represented by the Canadian NRC/DAF Darrieus type, soon to recommence test operations in the Magdalen Islands, the similar concept Sandia 17 meter diameter rotor unit and the 5.5 meter diameter rotor Fokker machine. A hybrid approach is shown by the Dornier unit with its mixture of Darrieus and Savonius rotors, while the variable geometry VAWT is illustrated by the design developed by Reading University. Finally, the Giromill is included as representing the only vertical axis design to employ variable pitch blades.

All these units are the possible forerunners of much larger machines and are therefore of interest to a utility.

The design state-of-the-art was reviewed and it became apparent that the HAWT is more advanced than the VAWT and that different design philosophies are represented by the various machines existing today. No clear indication exists as to which concept will produce the most cost effective system. However, it is clear that the present-day designs show that large WECS are technically feasible and that in a wind regime of, say 8m/sec mean wind speed, at 30 m height, a system can be made to be cost

effective when compared with equivalent electrical energy obtained from diesel oil.

One of the operating features necessary for utilities will be the capability of a WECS to operate throughout all types of weather, including icing. It is noted that none of the machines listed has the capability of deicing the rotor blades.

Reviewing the spectrum of WECS existing today or planned for the next two or so years, it is clear from a utility viewpoint that the technology has not developed to the point where a utility could write a procurement specification to which a manufacturer could respond on a production basis.

Existing constant speed machines, directly connected to the electrical network do not fully meet utility requirements from the viewpoint of quality of power generation. However, with utility company participation it appears feasible to design machines which will meet these requirements.

Figure 6 presents the major events of the last six years and a projection as to when the HAWT's and a VAWT will have demonstrated design reliability and performance.

This suggests that by the early to mid 1980's it is possible that one or more production systems could be available, which, with suitable influence by the utilities during the design stages, could be on site and operating so that wind power could begin to pay its way and make a fair return in the utility marketplace whilst extending to a considerable amount the use of non-renewable energy.

When considering the economics of owning and operating large size WECS, the following factors have to be taken into account.

1. Basic design of the machine
2. Suitability of the design with respect to:
  - a) wind regime at potential sites
  - b) employing electrical or equivalent storage facilities
  - c) integration into the existing grid system
  - d) application to the wind farm concept
3. The WECS design power rating and the projected electrical power production in kwh for the potential sites
4. Wind site data
5. Site preparation costs
6. Installation costs including site access
7. Maintenance and operating costs
8. Long life (minimum 30 years) with minimum replacement costs.

With the above factors optimized or near optimized for the 30 years system life, application of wind power to the production of electricity by utility companies can become economically attractive. The key aspect, it is felt, is to involve utilities, consultants and the manufacturing industry in the entire process of design.

### CONCLUSIONS

- 1) There is a growing interest in the Western European countries and North America in the WECS and their utilization for the generation of electrical power.
- 2) HAWT are more advanced in design and in operating experience than VAWT units. Both concepts show promise of technical and economic viability but, as yet, there are no clear cut conclusions as to which is the better concept.
- 3) Results of the various studies examined indicate that the present-day designs of large WECS are technically feasible and that in an 8 m/s meanwind speed appear capable of producing a cost effective system when compared with diesel units fuelled at present-day oil prices.
- 4) Existing constant speed machines, directly connected to the electrical network do not fully meet utility requirements from the viewpoint of quality of power generation. However, it appears feasible to design machines to meet the utility requirements.
- 5) The present-day designs appear to have a grid penetration limitation of about 10 to 15 percent without additional energy storage. Utility interest in WECS depends upon the amount of grid penetration capability of the WECS design.
- 6) The present-day designs of the WECS are still, in our opinion, preliminary designs and require design and cost optimization to be suitable for utility acceptance.
- 7) Based on the survey of the present-day development programs, it appears that the potential of large WECS will be established in the period 1983 - 1985.

## DISCUSSION

- Q. What is the height, power and diameter of the chosen design?
- A. The height of the chosen design was 100 meters from the ground. The distance from the bottom bearing to the top bearing was 96 meters and the rotor diameter was 64 meters. The power was 3,900 kilowatts, and the output will be on the order of 7,500,000 kWh per year.
- Q. I talked to a young man from Quebec in May that was, I think, actually operating a three-bladed 30-foot diameter horizontal axis machine. Could you give me some information on what's going on in that area of work?
- A. The gentleman concerned, Mr. A. Watts of Hydro-Quebec is immediately in front of you, and he has more information on that than I have.
- Q. Could you discuss the control of these machines?
- A. The large vertical machine which we looked at will be grid-controlled with a constant rpm locked into the grid. So the actual control would be the normal control of the Darrieus turbine with a given rpm.
- Q. If you had large installations of these, say, of five to a hundred megawatts, is that type of control accessible to your utility?
- A. Yes, certainly. That would be the way we would propose to go, because we have a very strong grid in the province of Quebec, and we don't see this machine replacing hydro power very much. We would regulate the rpm by synchronizing with the grid. I agree there could be a lot of discussion about the best way to do it, but that is the way we see it at the present time.

TABLE I. - HYDRO-QUEBEC

OWNERSHIP: Government-owned utility

TYPE OF GENERATION: 99.7% Hydro 0.3% Others

SYSTEM PEAK IN 1979: About 16,000 MW

TRANSMISSION SYSTEM: 25,000 km of circuits rated  
between 69 - 735 kV.

R & D: L'Institut de Recherche de l'Hydro-Québec (IREQ)

REASONS FOR INTEREST IN WIND ENERGY:

- Large potential for wind energy (55,000 MW)
- Reduce dependence on imported energy forms
- By 1990 most of the potential Hydro sites  
will have been exploited.

TABLE II. - WTG SURVEY

NAME	COUNTRY	STATUS (late 1978)	DESCRIPTION		
			Type	Blades	Rating, kw
Gedser*	Denmark	Test	Upwind HAWT	3	200
Mod A	"	Proposal	" "	3	630
Mod B	"	Proposal	" "	3	630
Tvind*	"	Test	Downwind HAWT	3	1724
Dornier*	W.Germany	Test	VAWT Darrieus & Savonius	3 - 2	20
GROWIAN	"	Design	Downwind HAWT	2	2000-3000
Fokker*	Netherlands	Test	VAWT Darrieus	2	15
FDO	"	Design	Upwind HAWT	2	300
Saab-*	Sweden	Test	Downwind HAWT	2	63
Scania					
Swedish	Sweden	Proposal	Downwind HAWT	2or3	2000-4000
Gov't					
Lawson-	UK	Test	Upwind HAWT	3	30
Tancred					
WPG (a)	UK	Design	Upwind HAWT	2	3700
Reading*	UK	Test	VAWT	2	1
Univ.					
Mod 0*	USA	Test	Downwind HAWT	2	100
Mod 0A	USA	Test	Downwind HAWT	2	200
Mod 1	USA	Installation	Downwind HAWT	2	2000
Mod 2	USA	Design/ Manufacture	Upwind HAWT	2	2500
WPPC*(b)	USA	Design/ Manufacture	Upwind HAWT	3	3000
WTG	USA	Test	Upwind HAWT	3	200
Sandia	USA	Test	VAWT Darrieus	2or3	60
Giromill	USA	Design	VAWT	3	40
NRC/DAF*	Canada	Test	VAWT Darrieus	2	230
Cdn.Gov't.Canada		Proposal	VAWT Darrieus	2or3	3800

\* Machines visited

(a) Wind Power Group

(b) Wind Power Products Corp.

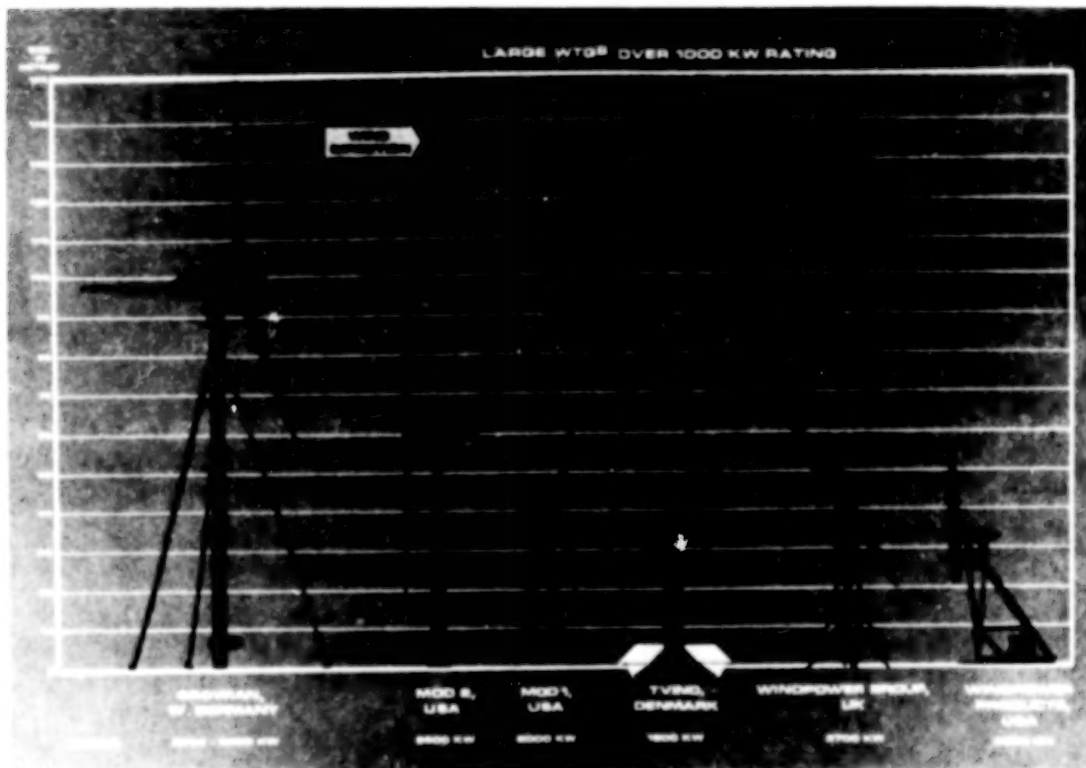


Figure 1

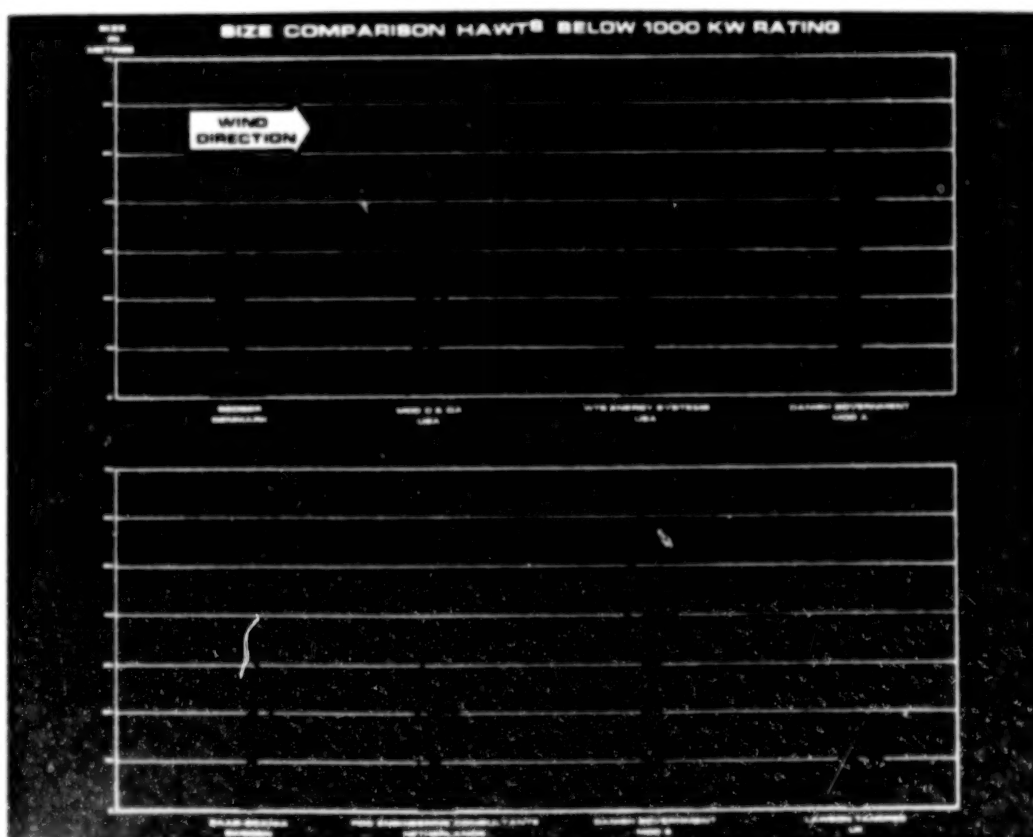


Figure 2



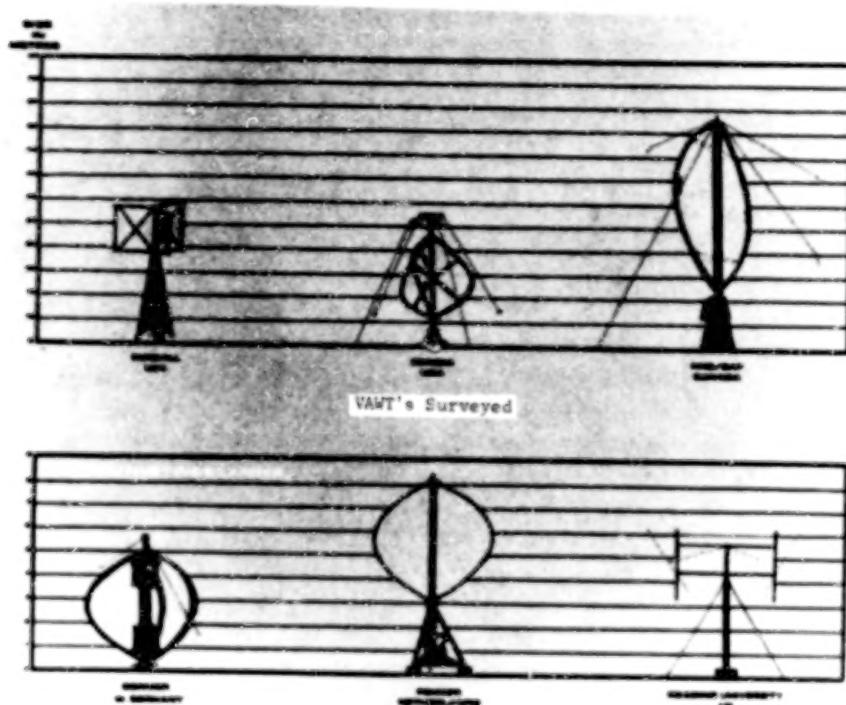


Figure 3

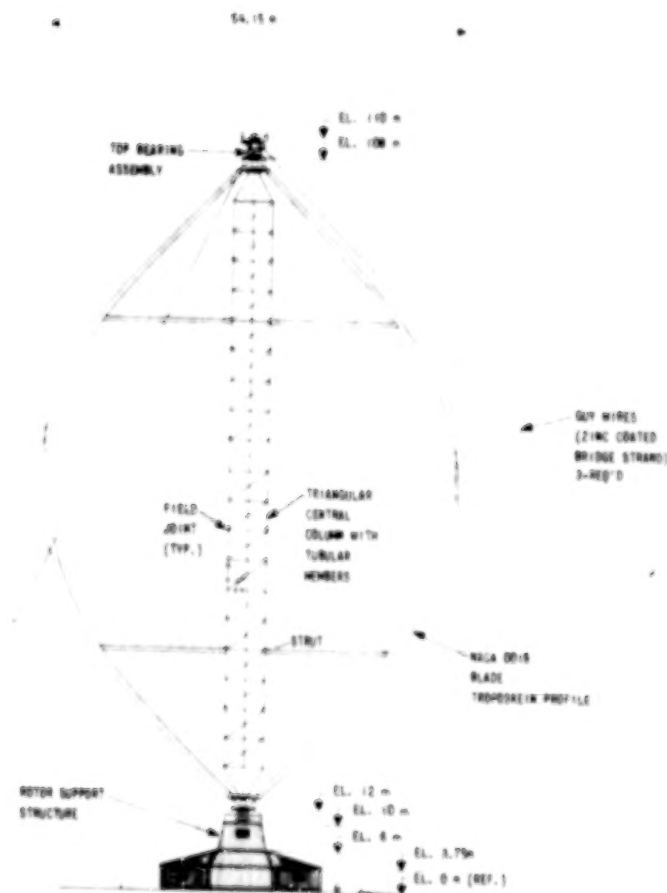


Figure 4. - Proposed Large VAWT.



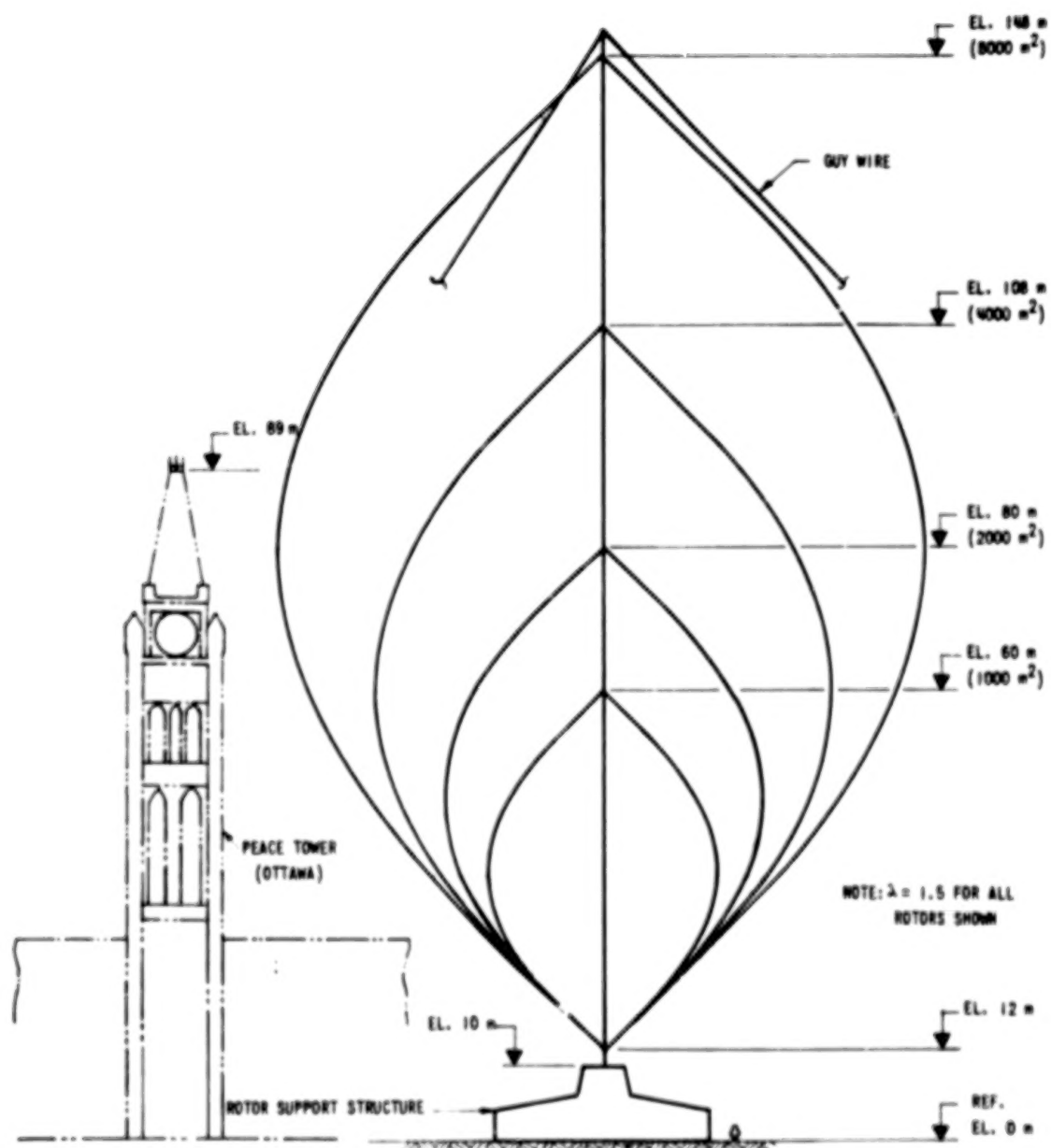
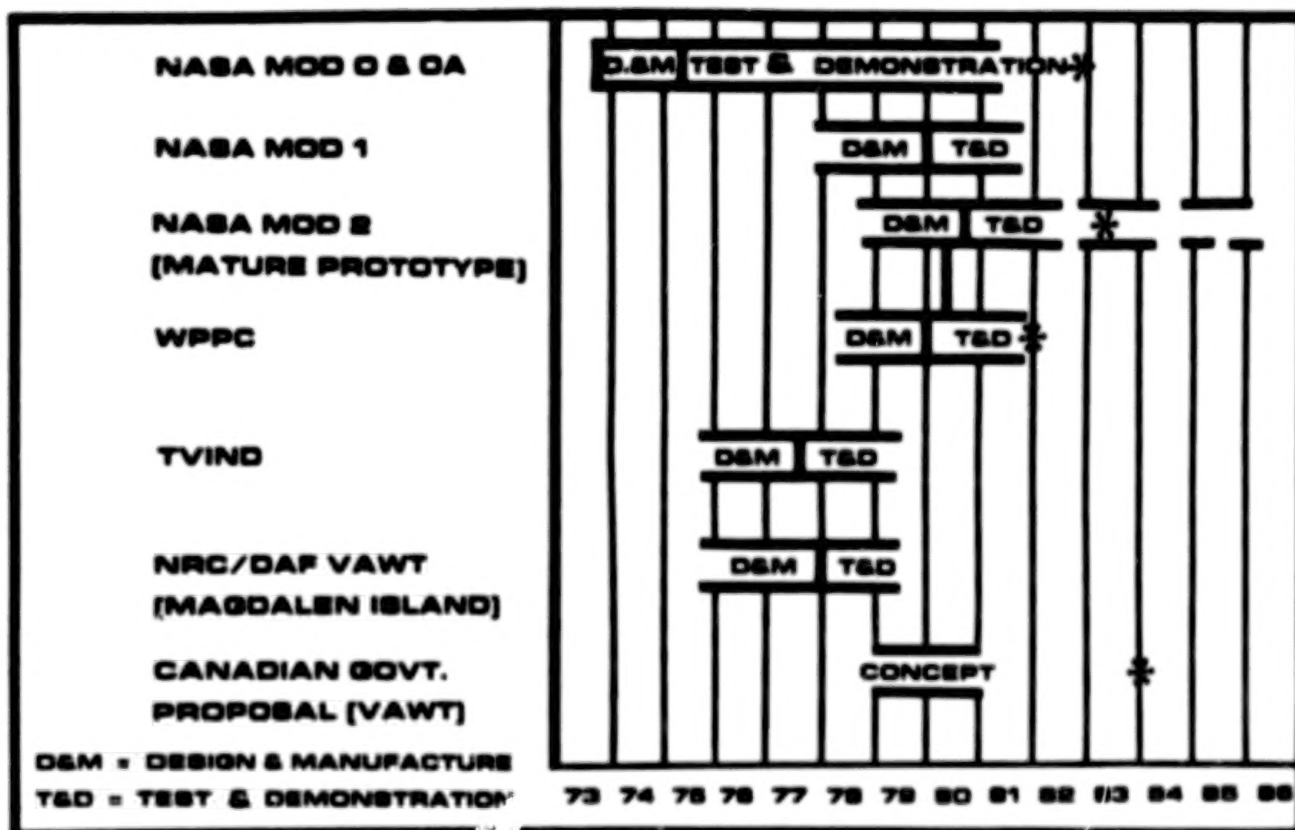


Figure 5. - Rotor size for 1000, 2000, 4000 and 8000 m<sup>2</sup> swept area.



\* = Anticipated date for demonstration of design reliability and performance.

Figure 6. - Recent Major Events.

## SIMULATION STUDIES OF MULTIPLE LARGE WIND TURBINE GENERATORS ON A UTILITY NETWORK

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### SUMMARY

The Wind Energy Project Office of NASA-Lewis has undertaken, as part of the Department of Energy wind program, a multi-faceted project to study the anticipated performance of a cluster of wind turbine generators on an electric utility network. Both in-house and contractor resources are being used. Preliminary results of an in-house simulation of two Mod-2 systems tied to an infinite bus indicate favorable system performance.

### INTRODUCTION

With the impending construction in 1980 of three Mod-2, 2.5 Megawatt, wind turbine generator systems, the necessity to identify any potential problems inherent in the operation of a cluster of such machines on a utility network becomes critical. The nature of wind turbine torque and drive train characteristics coupled to the variability of the wind excitation is such as to produce a generating system that warrants special analyses to assure its compatibility with established electric utility networks. In accordance with this requirement, the Lewis Research Center of NASA, as part of the DOE wind program, has established a project to investigate the interaction between a cluster of large wind turbine generators and a utility network.

The effectiveness of a single large wind turbine generator (WTG) synchronized to an electric utility network has been demonstrated by the Mod-0A, 200 kw, wind turbine generators operating at Clayton, New Mexico and Culebra, Puerto Rico, and the 100 kw, Mod-0 generator at the Plum Brook Station of the Lewis Research Center. The analyses of these machines on their networks have been reported by Seidel (Ref. 1), Reddoch and Klein (Ref. 2), and Power Technologies, Incorporated (Ref. 3).

Although concern for the possibility of interaction problems between wind turbine generators and a network has been expressed in the past, few results of studies of this interaction have been published. Pantalone and Fouad (Ref. 4) have studied multi-unit operating point stability with a constant

wind excitation, a parametric eigenvalue analysis.

The General Electric Company performed a study for ERDA (Ref. 5) which initiated an investigation into the interface design requirement for large scale application of wind turbine generators. Currently the Division of Electric Energy Systems of the Department of Energy is sponsoring a study of the dynamics of multi-unit wind turbine generators on electric utility systems. This study is being performed by Power Technologies, Incorporated and is scheduled to be completed by January 1980.

The intent of this paper is to describe the approach of the Lewis Wind Energy Project Office to developing analysis capabilities in the area of wind turbine generator-utility network computer simulations and to present some preliminary results of an analog simulations study of multiple-unit wind turbine generators tied to a utility network.

#### MOD-2 SIMULATION

The simulation model of the Mod-2 wind turbine generator that has been modeled in the hybrid computer laboratory at the Lewis Research Center is shown in Figure 1. The Mod-2 is a horizontal axis system with a two-bladed, upwind rotor driving a 2.5 Megawatt synchronous generator. Among the distinctive features of the system are the low-speed quill shaft designed to reduce torque oscillations by virtue of its relatively high compliance, and a two-per-revolution notch filter in the actuator command signal path to relieve the actuator from cycling at that frequency. The effects of both of these features are apparent in the simulation output signals.

The objective of this paper is to demonstrate the nature and application of the simulation. Detailed parameter values are not included here other than to indicate that they were obtained from the Boeing Company. Implementation of the simulation was completed in February and only limited studies of the type illustrated here have been performed. The data used was that included in the "Mod-2-107 Preliminary Design Blade Pitch Control System Simulation Report" by the Boeing Company, dated February 27, 1979. As the final design proceeds and parameter values are better defined, the simulation will be modified to reflect the latest available parameter values.

Analog Simulation of a Two-Wind Turbine System. - The simulation of two Mod-2 wind turbines, radially connected to an infinite bus, was implemented through a joint effort of Purdue University and NASA-Lewis. The common radial connection of the two generators provides an electrical interaction between them.

It was determined at the outset that representation of electromagnetic transients in the radial transmission line and in the machine stators would not be required, since such transients are not important to electromechanical

motion. Thus, the electrical representation of each generator involves three state variables, namely field flux linkage and damper flux linkage in the q and d axes. Use of flux linkages as state variables allows a convenient and physically reasonable representation of saturation.

Torsionally-induced motion of the system is represented by three rotational inertias (1. blades, 2. hub and quill shaft, 3. gearbox and generator) connected by elastic shafts. Six state variables are involved: three speeds, two shaft torques, and the displacement between the generator inertia and a constant angular velocity reference.

A brushless exciter produces field current for the generator. Simulation of this component is done with only one state variable (exciter field flux), plus several algebraic relations to describe the action of the rotating rectifier. The static voltage regulator model adds three more state variables.

The simulation of aerodynamic effects is done by algebraic equations. Wind torque is proportional to a power coefficient and to the cube of the wind velocity, and is inversely proportional to rotational speed. Curve fitting is used to represent the power coefficient as a function of tip speed ratio and pitch angle.

The pitch control subsystem in the simulation is that designed by Boeing. It uses four state variables: one in an integral controller, one for the hydraulic servo, and two for a "2P" notch filter. In all, seventeen state variables are employed in each wind turbine model.

The equations and integrations required for one wind turbine are implemented on an EAI 681 analog computer. Simulation speed is real time or ten times slower than real time. Two EAI 681 consoles are operated in parallel to implement the two wind turbine system. The simulation is in real time and interactive. Repeated runs with parameter changes are quickly accomplished.

Comparison with Boeing Simulation Results. - The first response examined the response of a single wind turbine generator, rigidly tied to an infinite bus, to a  $(1-\cos)$  gust. This response was examined to compare with results obtained by the Boeing Company, the designer of the Mod-2 system, with its digital simulation of a single machine (Ref. 6). The results of the two simulations matched closely. The comparison figures are included in Figure 2. The steady-state wind speed was 28 mph. Upon this was superimposed a 30-second  $(1-\cos)$  gust which produced a peak speed of 1.41 times the steady-state speed. As a result of the 41% gust, each model indicated  $\pm 8\%$  variation in electromagnetic torque and a pitch angle change of 10.5 degrees. The NASA model showed a power angle change of  $\pm 2$  degrees; the Boeing model,  $\pm 1.5$  degrees.

Gust Response of Two Machine Model. - The simulation was developed for two identical Mod-2 wind turbine generators. The two systems were coupled to an infinite bus through a reactive line as depicted in Figure 3.

The two coupled systems, each operating in a 28 mph steady wind, were subjected to a 13.8 second, 28% increase in wind speed,  $(1-\cos)$  gust. This gust has been designated the "Design Gust" for the Mod-2 wind turbine generators (Ref. 5). The response of each system to gusts at two different time relationships between the gusts is shown in Figures 4 and 5. The transmission line reactance was a relatively high 0.4 p.u.

Figure 4 shows the response of the two machines to a gust on machine #1 only. There is about a  $\pm 8\%$  change in rotor torque for machine #1, but the only change observed in machine #2 is a  $\pm 1$  degree change in the power angle.

In Figure 5 responses are shown for both systems subjected to the same gust, but machine #2 experiences the gust about 7 seconds after machine #1 does. No interaction is observable and each system appears to respond as it would without the presence of the other.

For the two sets of gust responses illustrated, the system is stable. There is little, if any, interaction of the quantities monitored for the particular operating point and configuration of generators modeled.

One WTG Tripped Off Line. - A case was run simulating the response of WTG #2 when WTG #1, which had been operating in parallel with WTG #2, dropped off the line. Both machines had been operating at rated load in a 28 mph wind, connected to an infinite bus through a 0.4 per unit reactive line. The resulting transient on WTG #2 lasts about 2 seconds, decaying rapidly. The machine remains synchronized and the output power returns to the rated value. This response is illustrated in Figure 6. It is pertinent to note that the model at this stage included no rate limit on the control blade pitching. Future simulation runs will include a rate limit and may produce different results.

Fault at WTG Bus. - With the two machines operating in parallel in a 28 mph wind and each generating rated power, a .04 second three-phase short circuit was applied at the common WTG bus. A local load of 2 per unit was also tied to the system at the WTG bus. The WTG bus was connected to the infinite bus through a 0.4 per unit reactive line. The results of this simulation is shown in Figure 7. The signal traces for only one machine are shown. The second machine traces are identical. The machines quickly recover from the effect of the fault when the portion of the network including the fault is removed. After a half-dozen swings, the transient has disappeared and the system has returned to its original condition.

Effect of Actuator Time Constant. - In order to observe the effect of doubling the time constant of the blade pitch actuators simulations of faults on the parallel WTG's were compared using the nominal time constant and a time constant of twice nominal value. The results are shown in Figure 8. With the two WTG's operating at rated load in a 28 mph wind a 0.02 sec three-phase short circuit occurs at the WTG bus. In each case, when the fault is cleared, the systems return to the pre-fault operating condition. (Traces of only one machine are shown.) Close inspection of the traces show the effect of the longer time constant. However, the effects are very small, and considerable variation of the time constant parameter appears to be possible.



### CONCLUDING REMARKS

The Lewis Research Center Wind Energy Project Office has undertaken a multi-faceted investigation of potential electrical problems that may be inherent in the intertie of clusters of wind turbine generators and an electric utility network.

Preliminary and limited results of the analog simulation study of two Mod-2 wind turbine generators tied to an infinite bus indicate little interaction between the generators and between the generators and the bus. The system demonstrated transient stability for the conditions examined.

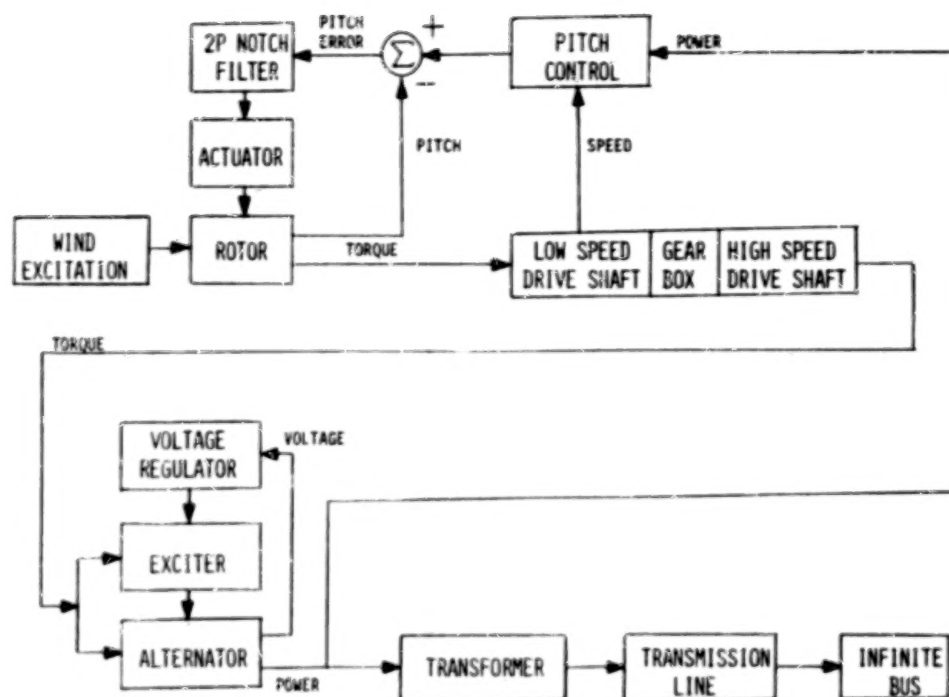
### REFERENCES

1. Seidel, Robert C.; Gold, Harold; Wenzel, Leon M.: Power Train Analysis for the DOE/NASA 100-kw Wind Turbine Generator. NASA TM-78997, October 1978.
2. Reddoch, T. W.; Klein, J. W.: No Ill Winds for New Mexico Utility. IEEE Spectrum, Vol. 16, No. 3, March 1979.
3. Hannett, L. N.; Undrill, J. M.: Wind Turbine Operation in Parallel to Diesel Generation. Power Technologies, Inc., Report No. R-42-77, August 1977.
4. Pantalone, D. K.; Fouad, A. A.: Modes of Oscillations of Wind Generators in Large Power Systems. Paper A78 579-5, IEEE Summer Power Meeting, Los Angeles, July 1978.
5. General Electric Company: System Dynamics of Multi-Unit Wind Energy Conversion System Applications. Final Report, General Electric Space Division, September 1977.
6. Boeing Company: Mod-2-107 Preliminary Design Blade Pitch Control System Simulation Report. The Boeing Company, February 1979.

## DISCUSSION

- Q. Serious problems usually occur as a result of more than one fault happening simultaneously. Are you planning to look into two or three things happening at the same time?
- A. Yes. These results were obtained last week, so they are preliminary. There are many, many things to study, and the ones you suggest certainly are pertinent.
- Q. You have said that the inertia constant is high. What is the inertia constant for the Mod-2?
- A. It is 16.5 kilowatt seconds per kva.
- Q. You mentioned that the control system was very rapid. What was your time constant?
- A. I believe the time constant is one-sixth of a second.
- Q. I wasn't sure exactly what was covered in the Power Technologies study. When are the results going to be available? Secondly, General Electric did make a study of power group stability. Are you familiar with that report, and do you know what it contains? Is it available yet?
- A. The General Electric study is not available. I think it is fair to say that it is not an extensive study. It was really an introduction to the problem, and GE ran out of time before really getting into the problem. Power Technologies Incorporated started with a one-year study for the Department of Energy in December of 1978. It is going to look at a single machine on a network and then two machines on a network. The parameter changes principally are going to be in the size of the machine, whether it is 200 kw or three megawatt, to see if there is a connection between size and performance. They are also going to look at the same kind of thing that I looked at.





MOD-2 SIMULATION BLOCK DIAGRAM

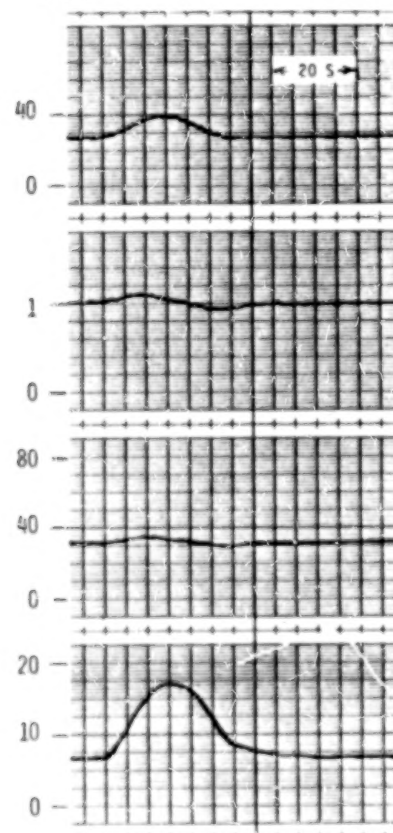
FIGURE 1

WIND SPEED, MPH  
GUST = 41%

ROTOR TORQUE, P.U.

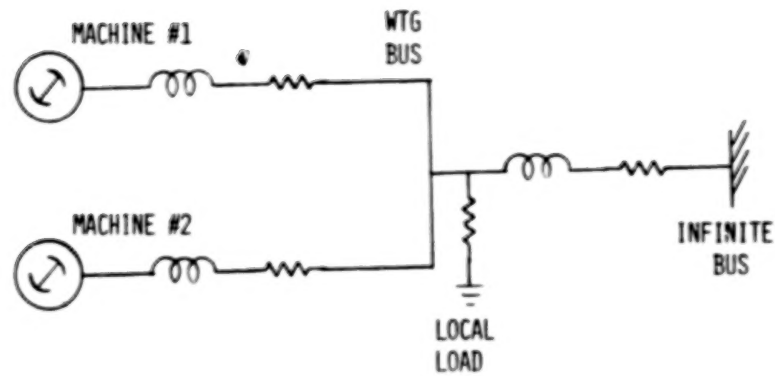
POWER ANGLE, DEG.

BLADE PITCH, DEG.



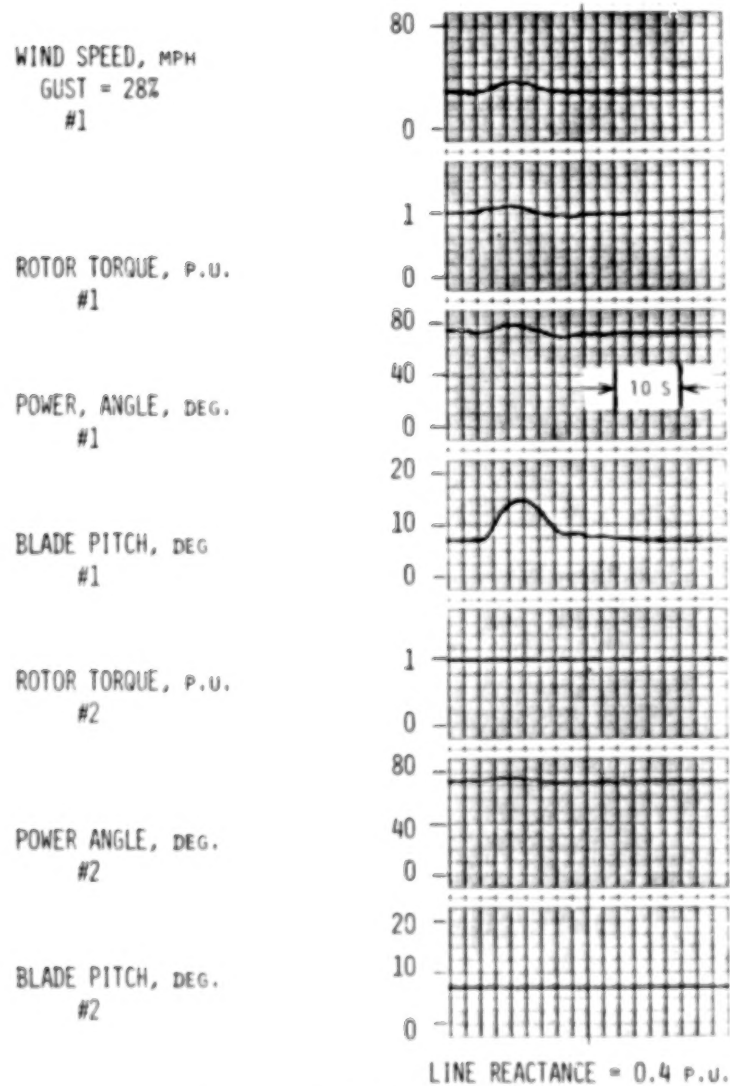
SINGLE MOD-2 LOAD GUST RESPONSE  
28 MPH + 41%, 30 SEC GUST

FIGURE 2



TWO WIND TURBINE GENERATORS ON AN INFINITE BUS

FIGURE 3



TWO WIND TURBINE RESPONSE

GUST HITS #1

FIGURE 4

WIND SPEED, MPH  
GUST = 28%  
#1

ROTOR TORQUE, P.U.  
#1

POWER ANGLE, DEG.  
#1

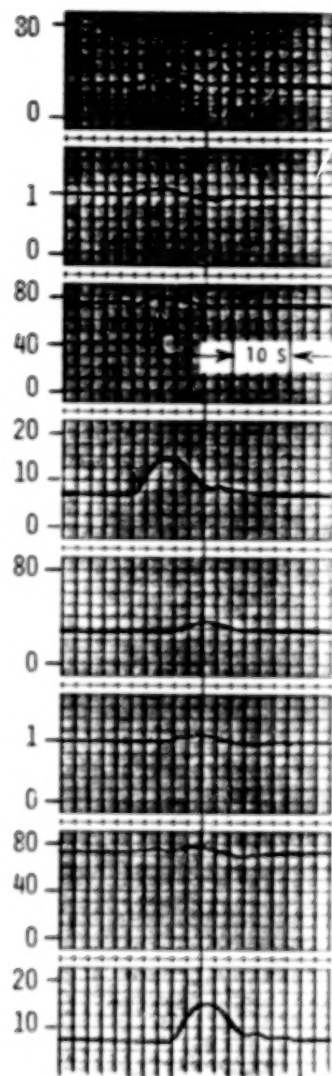
BLADE PITCH, DEG.  
#1

WIND SPEED, MPH  
#2

ROTOR TORQUE, P.U.  
#2

POWER ANGLE, DEG.  
#2

BLADE PITCH, DEG.  
#2



LINE REACTANCE = 0.4 P.U.

TWO WIND TURBINE RESPONSE  
GUST HITS #1 AND #2

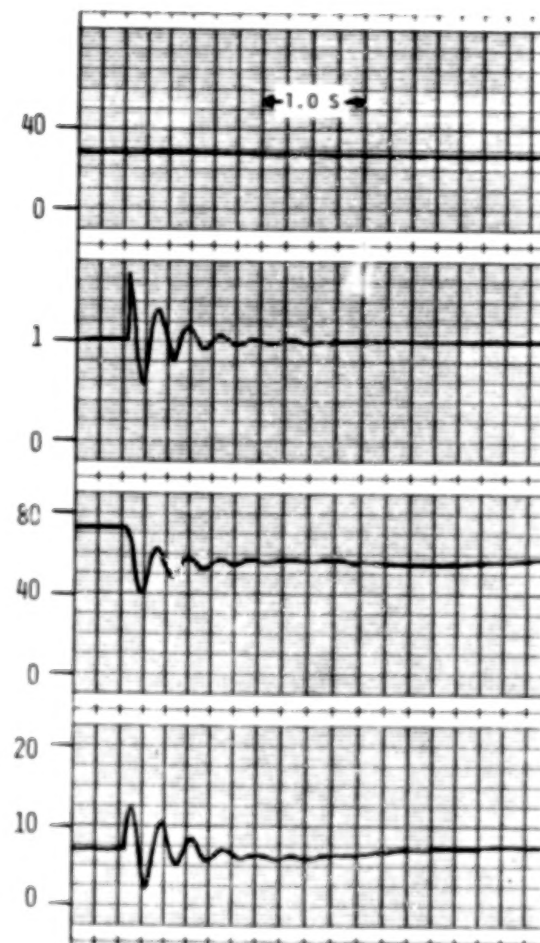
FIGURE 5

WIND SPEED, MPH

OUTPUT POWER, P.U.

POWER ANGLE, DEG.

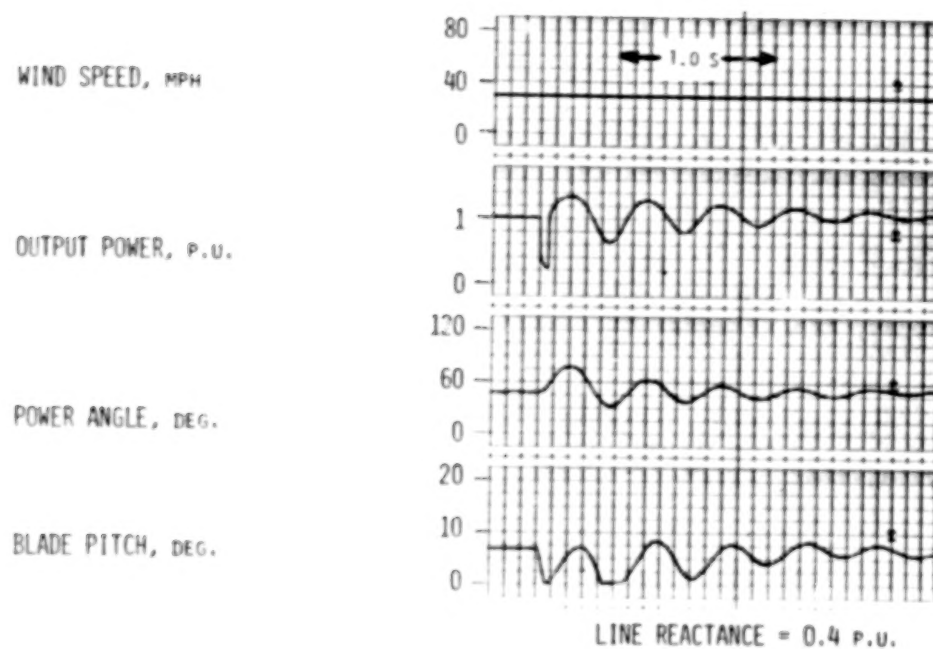
BLADE PITCH, DEG.



LINE REACTANCE = 0.4 P.U.

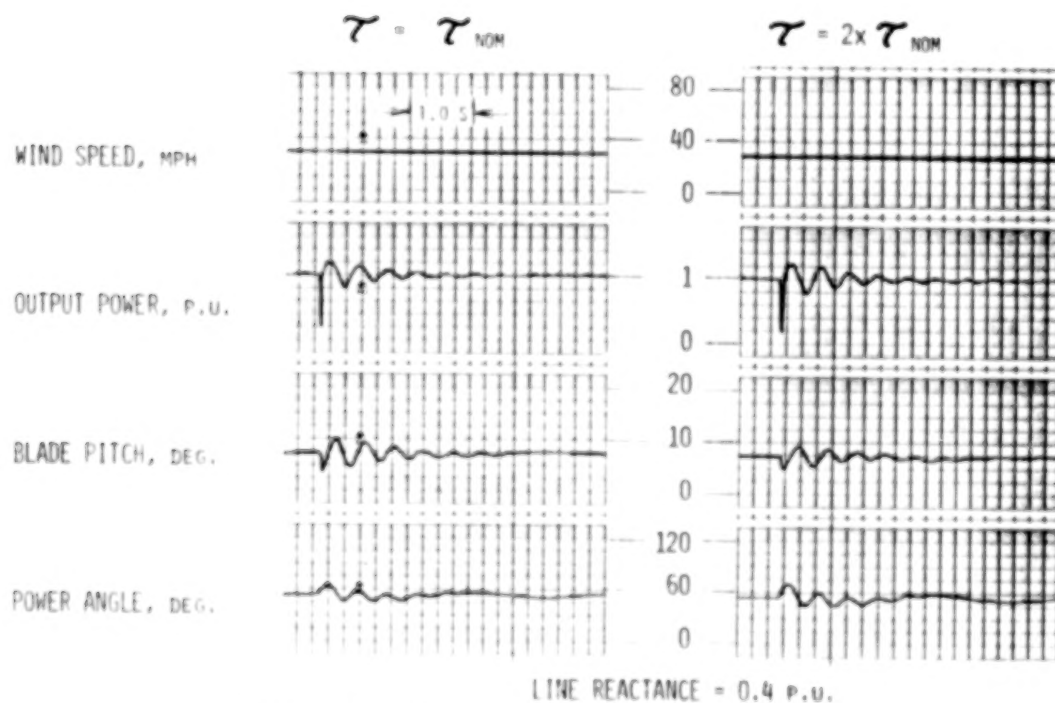
RESPONSE OF WTG #2  
WHEN WTG #1 IS TRIPPED OFF LINE

FIGURE 6



PARALLELED WIND TURBINE GENERATORS ON A NETWORK  
 .04 SECOND FAULT AT WTG BUS WITH LOCAL LOAD = 2 P.U.

FIGURE 7



PARALLELED WIND TURBINE GENERATORS ON A NETWORK  
 .02 SECOND FAULT AT WTG BUS  
 ACTUATOR TIME CONSTANT  $\tau$  VARIED

FIGURE 8

## SYSTEM CONFIGURATION IMPROVEMENT

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The design of a wind turbine generator is a very complex process because of the many (and often conflicting) choices and considerations involved. As a consequence, the determination of a superior system can best be achieved from intensive studies and probings that reflect the truly pertinent governing factors. This paper presents a discussion of such a process in terms of word charts and associated figures.

Factors involved in the choice of the system configuration are listed below. It has been found that choices among the many configuration options can be based strictly upon the resulting cost of energy results. Choices made on that basis also lead to reduced analytical complexity, less hardware complexity and reduced program risk. It was also found that many seemingly minor details turn out to have important impacts that are seen only after design, performance and cost-finding have been thoroughly probed.

The final result of these processes was the identification of a currently superior system, as indicated in the chart. The ensuing charts will examine the considerations that lead to this determination.

## THE CHOICE OF CONFIGURATION

- All Choices Can Be Made on Cost of Energy Basis
  - The Impact of Dynamics Configuration Can Be Large
  - There Are No Analytical Problems
  - Intensive Study of Capital Cost and Energy Capture Impacts Are Required
  - Design and Cost-Finding Refinement Can Change Early Judgements
- 
- Present View of Superior System
    - Two-Bladed, Teetered, Gravity-Balanced, Downwind Rotor
    - Yaw Free with  $\Delta_3$  to Correct Heading Trim
    - Soft, Tall Tower
    - Softened and Damped Drive
    - Full Span, Active Pitch Control
    - Actuators in Rotating System

Systems studies by Hamilton have spanned a wide array of concepts and dynamics design philosophies. We have kept pushing configuration improvement in all promising directions, rather than stop upon merely substantiating that a concept is feasible or satisfactory. At the same time, the behavior of unsatisfactory systems has been intensively studied to gain an understanding of cause and effect aspects and to build confidence in our computer codes.

Analysis in a time domain is of greatest engineering value because it displays both transient and steady state loads and stability conditions. System behavior can be studied, modified, and improved. Our F-762 program has had the benefit of validation during full-scale helicopter flight and wind tunnel tests and also for wind tunnel tests of scale model wind turbines. Although F-762 is still evolving in its detail features, it has become established as a very adequate and reliable design tool. It can handle wind turbine configurations that are much more difficult than those we are finding to be superior.

The dynamics of wind turbine generator systems exerts a large impact on the cost of energy for the system. Dynamics considerations can decrease capital costs and increase energy capture. System dynamics is also the key factor in the quest for unlimited fatigue life and increased utilization.

The major basic configuration options that were considered in the system dynamics trade-offs, and the resultant selected configuration, are listed below.

## **BASIC CONFIGURATION TRADE-OFFS IN SYSTEMS DYNAMICS**

### **Stiff Vs. Soft Structural Design**

### **Trimmed Vs. Untrimmed Vibratory Airloads**

#### **Configurations Examined:**

- Rigid Rotor on Stiff Tower with Stiff Drive System — First Generation**
- Rigid Rotor on Soft Tower**
- Cyclic Pitch Rotor on Stiff Tower**
- Teetered Rotor on Stiff Tower**
- Teetered Rotor on Soft Tower**
- Soft Drive System with Torsional Damping**

---

#### **Selected Configurations:**

- Teetered Rotor on Soft Tower**
- Soft Drive System with Torsional Damping**

#### **Why:**

- Large Reductions in Operating Loads**
- Makes Soft Tower Feasible**
- Simplifies Rigorous Analysis**
- Soft Drive System Keeps System on Line**

One of the earliest design considerations is the form of the blade articulation. The chart below lists the three options studied, with resultant major advantages and disadvantages for each case.

The choice of a teetered (two-bladed) rotor configuration has large advantages and no significant disadvantages. When combined with a slender, soft tower and with soft drive system concepts, the teetered rotor presents such low vibratory load levels that there is no remaining reason for choosing an upwind rotor location. Unlike either hingeless or individually flap-hinged configurations, the teetered system can be so balanced that it achieves full, maximum energy capture when operating in a shear gradient.

For free-yaw operation the teeter hinge can be modified to correct a heading trim error that would sacrifice energy capture performance. The subject trim error is characteristic of all three blade articulation concepts when operating in a shear gradient.

## **FORM OF BLADE ARTICULATION**

### **● Hingeless**

- Very Costly Vibratory Loads
- Yaw Rate and Yaw Angle Are Limited
- Cannot Be Positioned for Maximum Energy Capture

### **● Individual Flap Hinge**

- Free of Flatwise Root Bending Loads
- No yaw Rate or Yaw Angle Limits
- Loss of Energy Capture Due to Excess Coning
- Inevitable Gravity Induced 2P Vibration
- Inevitable Gravity Induced Power Loss

### **● Teeter Hinge**

- Free of Odd Integer Flatwise Root Bending
- No Yaw Rate or Yaw Angle Limits
- Coning Can Be Restrained
- Can Be Gravity Balanced
- Will Trim Itself for Maximum Energy Capture



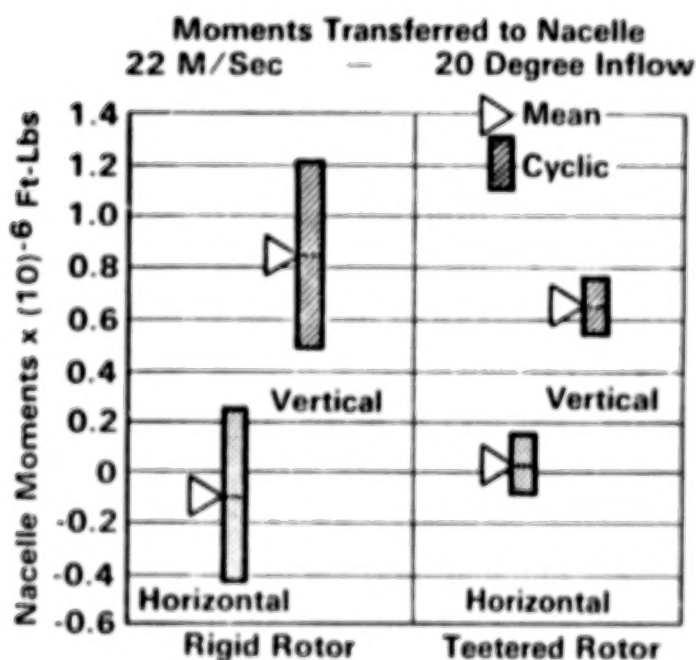
Reduction of system vibratory loads is the cumulative result of numerous choices available to the designer. As indicated previously, the form of blade articulation is a major factor.

The plot below shows the reductions of nacelle shaking moments that can be obtained from merely introducing a teeter hinge at the apex of the rotor cone. For this location, the residual vibratory moments are significant and of a two-per-rev frequency.

However, by placing the teeter hinge on the rotor center of gravity, these two-per-rev moments are reduced substantially to zero. Surprisingly, when the rotor is thus balanced, the energy capture also improves.

With the final balanced rotor, the only vibratory loads reaching the nacelle are small two-per-rev thrusts that exert only  $\pm 0.02$  g on the nacelle. Tower vibratory stresses drop to only 2% of steady, thus eliminating fatigue as a design problem.

## NACELLE AND TOWER LOADS REDUCTION





A second major design choice is the stiffness of the drive system and tower. Some major considerations regarding stiffness are listed in the chart below.

After thorough system dynamic study, it becomes apparent that there are very few aspects of wind turbine design that benefit from high stiffness. Stiffness is only essential and valuable to protect the blades against resonant first mode bending response to gravity loads. The choice of low stiffness design concepts directly attenuates the vibratory structural loads and improves the quality of the power delivered.

Low stiffness requires detailed attention to matters of system stability. However, solutions are found to be simple and straightforward.

## **SOFT VERSUS STIFF DESIGN CONCEPTS**

### **● Where Stiffness is Useful**

- First Edgewise Blade Mode to Preclude Gravity Resonance
- First Flatwise Blade Mode to Resist Coning and to Avoid Gravity Resonance if Blade is Fully Feathered

### **● Where Stiffness is Detrimental**

- In Drive System It Precludes Benefits of Uncoupling the Rotor Inertia
- In Tower It Moves Coupled Frequencies Up to Create Resonance Problems
- In Blade Modes Above First It Inhibits Damping and Increases Bending Moments

### **● Stability Aspects**

- Modal Stiffness May Be Tailored to Assure Stability But High Stiffness Per Se is Not Required

### **● The Value of Low Stiffness**

- Reduced Vibratory Loads
- Steady Power Output
- Reduced Weight

The major considerations involved in the question of fixed or variable pitch for the rotor are delineated in the chart below. Variable pitch can be either part span or full span.

Choice of partial span fixed pitch is of interest and feasible from the standpoint of system dynamics and of rotor aerodynamic performance. However, system studies to date indicate that partial span pitch leads to tower costs that tend to inhibit tower height increase. This in turn leads to poor COE performance in an environment that has a significant shear gradient.

Thus it appears that full-span pitch control will usually deliver best system COE performance.

## **FIXED VERSUS VARIABLE PITCH**

### **● Value of Fixed Pitch**

- Simplicity in the Rotor
- First Flatwise Blade Frequency Can Be Reduced

### **● Obstacles to Achievement**

- Starting and Overspeed Control Not Provided
- Energy Capture is Reduced in Synchronous Systems
- Machine Comes Closer to Stall
- Hurricane Loads Become Greater Problem
- Higher Thrust Loads Lead to Limited Tower Height

### **● Prospective Compromise**

- Partial Span Control Can Remove First Flatwise Frequency Problem
- Energy Capture Suffers Directly
- Tower Height Compromise Also Loses Energy Capture
- No Problem with Outboard Actuator Mass

The question of degree of yaw control is addressed in the table below. The two options considered are active yaw control or complete yaw freedom.

Provided the system is given stable and correct heading trim characteristics when operating downwind, there appears to be no reason to adopt an active yaw control. The trim correction can be provided in teetered or individually flap-hinged configurations. Hingeless systems cannot be given the necessary stable heading trim behavior without adding complex corrective devices.

## **YAW CONTROL VS. YAW FREEDOM**

- **Value of Yaw Control**
  - Needed Only if Rotor is Upwind
  - Useful for Service Orientation
- **Disadvantages**
  - Consumes Parasite Power
  - Depends Upon Heading Sensors
  - Adds Capital and Maintenance Costs
  - Introduces Tower-Coupled Yaw Modes
- **Essentials of Yaw Free Operation**
  - Downwind Rotor
  - Stability
  - Inherently Correct Heading Trim
  - Feathered for Pre-Start
  - Vertical Parking

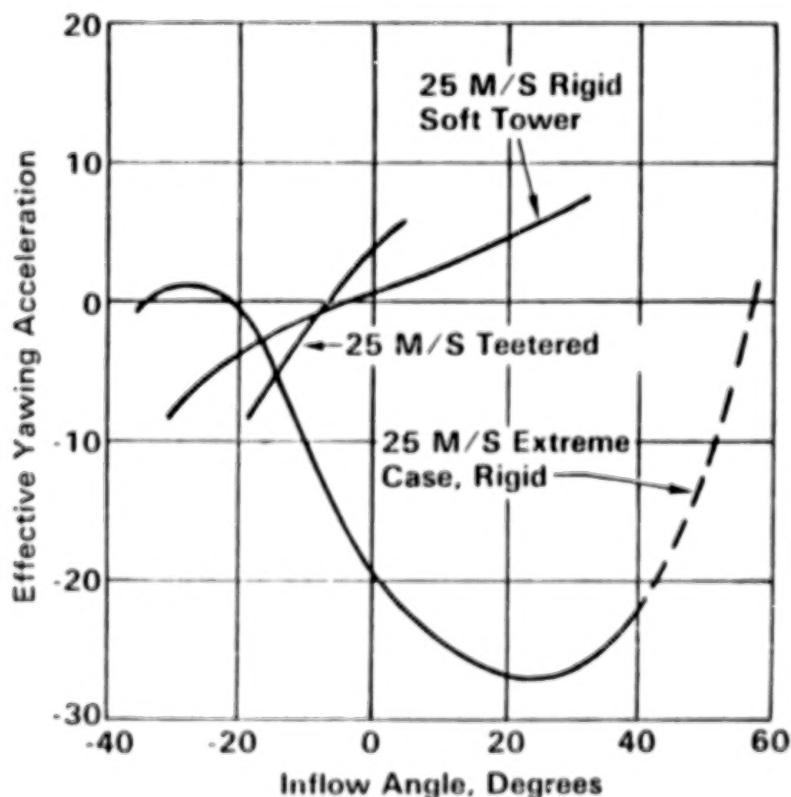
The characteristics of free yaw systems for several rotor configurations are shown in the plot below. Teetered and hingeless rotor are compared.

Both adverse yaw trim and heading instability are characteristic of hingeless free-yaw systems. As seen in the lower curve, a hingeless system under the influence of a torque limiting pitch control (well above rated power) has two stable trim positions - one at 35° left and another at 55° right yaw. An unstable equilibrium also exists at 22° left. This totally unsatisfactory trim behavior is accompanied by large vibratory loads build-up in the right yaw sector.

By contrast, the teetered rotor displays a highly stable behavior.

Also plotted is the surprising behavior of the rigid rotor when mounted on a very soft tower. Tower elastic deflection provides a teeter hinge effect and stabilizes the yaw behavior! The cost, however, is high vibratory loads in both rotor and tower.

### HEADING TRIM AND STABILITY FOR FREE YAW SYSTEMS



The control of static and transient thrusts is discussed in the chart below. Static and transient thrusts have their impact on cost of energy through tower costs that inhibit tower height.

A softening of the tower has benefits analogous to a softening of the drive, in that system inertia is uncoupled to better absorb impulsive loads. As with the drive system, damping is needed to control transients. This is well provided in most modes by the inherently large aerodynamic damping forces that come from axial (thrustwise) motions of the rotor disk. Furthermore, torque control system design must also give attention to the thrustwise system modes and their responses to pitch change.

## **CONTROL OF STATIC AND TRANSIENT THRUSTS**

### **● Impact**

- Capital Cost of Tower Can Be Reduced
- Blade and Hub Fatigue Life Are Enhanced
- Tower Height Can Increase Energy Capture

### **● Feasibility**

- Benefit is Inherent with Torque Control
- Applies Also to Stand-Alone Machines Because of Gearbox Cost Limits
- Control Must Be Tailored to Avoid Adverse Effects in First Flatwise Blade Mode

### **● Related Issues**

- Rotor Thrust Damping is Large and Helpful
- Tower Softness is Helpful
- Stall As Thrust Limiter is Applicable Only to Synchronous Machines
- Result, Fixed Pitch Constant Velocity Ratio Probably Uneconomic

Good system damping is highly desirable. Considerations involved in system damping are listed in the table below.

It is found that systems that apply low stiffness design concepts are easily provided with good damping in all the pertinent modes. This in turn means that a given system can be run satisfactorily at various rotational speeds. Attention to achieving this capability can pay off in freedom to change operating speed with changes of site conditions.

## **ASSURANCE OF GOOD SYSTEM DAMPING**

### **● Impact**

- Ability to Run On or Across Rotor Resonant Frequencies
- Rapid Decay of Tower Bending Transients
- Rapid Damping of Torque Transients

### **● Feasibility**

- Flatwise Blade Modes Are Well Damped
- Tower Bending Modes Are Well Damped
- Damper on Gearbox Gets the Even-Integer Edgewise Blade Modes
- With Soft Tower the Rotor Thrust Damping Gets Most of the Odd-Integer Edgewise Blade Modes
- Damping Around the Yaw Bearing Can Get the Remaining Odd-Integer Blade Modes

### **● Related Issues**

- Ability to Adjust Operation to Various Average Wind Velocities
- Ability to Run at Variable Speed Under Non-Synchronous Loads
- Control System Design Requirements Are Eased

The attainment of unlimited fatigue life is an important objective for system design. The factors involved in this quest are presented in the chart below.

Virtually all of our dynamic configuration improvement choices have contributed to reducing vibratory loads and to easing the fatigue design task. Concurrently we have found that high wind, turbulent conditions can be accommodated without protective cutoff that would cost any significant energy capture. The choices that enable these gains have also been found to simplify the structure, eliminate subsystems, reduce weight and enhance energy capture.

## **ATTAIN UNLIMITED FATIGUE LIFE**

### **● Objectives**

- Avoid Premature Retirement
- Reduce Maintenance and Inspection
- Accomplish without Increased Capital Cost

### **● Approach**

- Reduce Applied Vibratory Loads
- Reduce Dynamic Response

### **● Methods**

- Eliminate First Harmonic Airloads with Teeter Hinge or Cyclic Pitch
- Uncouple Blade from Fixed System
  - a) For Flatwise Bending — Teeter Hinge
  - b) For Torque-Torsionally Soft Shaft
- Provide Multi-Modal Damping
  - a) Gear Case Damper
  - b) Nacelle Yaw Damper
- Provide Fast Pitch Control

## **CONCLUSION**

We have found that there is a large payoff from intensive effort to improve the system dynamics configuration. To apply the results most effectively, however, the design must evolve under constant and thorough scrutiny of the cost of energy impact of each choice.

## DISCUSSION

- Q. In conjunction with the rpm sweep from 25 to 85 rpm which passed through all of the resonances, what was the chordwise natural frequency at the high rpm?
- A. I don't recall a specific number, but the natural frequency remained well above one-per-rev. That model was run in an overspeed condition while searching for coupled mode instabilities on the soft tower. They were found around 150 rpm. The gravity edgewise resonance was well above 80 rpm. Also, we were very interested in whether yaw motion damping would be needed in the system. The conclusion, to our surprise, was that it wasn't.
- Q. Did you evaluate the possibility of having perhaps a selection of two or three frequencies as would be obtained from a gearbox or drive type of arrangement, instead of going with continuous variable tip speeds?
- A. That point is included in the study, but not very vigorously. We at Hamilton Standard are waiting now for the results of an electrical system study. My personal opinion at the present time is that it will be just as economical to have an electrical torque control which can hold a constant velocity ratio as any other way.
- Q. Would it be preferable for the blade to be on the stiff and lightweight side?
- A. As a previous comment pointed out, the blade has to be stiff enough edgewise to avoid the one-per-rev gravity resonance. It is not an important problem in the flatwise direction.



## COST OF ENERGY EVALUATION

Thad M. Hasbrouck  
Hamilton Standard  
Windsor Locks, Connecticut 06096

The cost of energy of a wind turbine generator system contains the three elements of Capital Cost, Operation and Maintenance Cost, and Energy Capture. In equation form, the Cost of Energy in cents/kilowatt hour is

$$\text{COE} = \frac{0.18 (\text{Capital Cost}) + \text{Levelized O\&M}}{\text{Kilowatt Hours Per Year}}$$

We at Hamilton Standard feel that this equation should be stenciled on the forehead of each of our designers, displayed prominently on office walls and, in fact, have the same emphasis as the THINK program at IBM.

Each of the elements of this equation is important and true. Low COE wind turbines will not become available until this is realized. Much emphasis has been placed upon reducing hardware capital costs, but if it is at the expense of increased O&M and reduced energy capture, we have accomplished nothing.

Much emphasis must be applied to reducing the indirect capital costs such as siting, foundations, and erection costs. The reduction in O&M costs is important because if we are not careful, this item can be a significant fraction of the numerator of the equation. Probably the greatest gains can be made in addressing the denominator of the equation - energy capture. Extremely careful attention to the one- and two-percent items that improve energy capture pay off in very significant reductions in COE.

The work that Glid Doman has described in his paper "System Configuration Improvements" is part of an extensive company-sponsored program to develop the concept of a wind turbine machine that Hamilton Standard feels is required for low energy costs. The results of this study in a machine we have named WTS-3 will now be presented.

An artists conception of the 3 MW WTS-3 wind turbine is shown in figure 1. The concept has a two-bladed rotor and a tubular tower. Major specifications and requirements of the WTS-3 are listed in figures 2 and 3. The power profile and yearly energy output variations are shown in figures 4 and 5, respectively.

The costs for the WTS-3 system were determined for 100 production units with a 14 mph wind at the 30 foot elevation. The results are:

INSTALLED COST - \$420 PER KILOWATT

ENERGY COST - 2.4¢/kWh

Such low estimated costs are in deed encouraging and can well pave the way for acceptable commercial utilization.

#### DISCUSSION

Q. The COE formula that you use, which was also presented by NASA, has been critized by many people. Could you give us your recommendation on how the costs should be calculated?

A. We have no argument with the formula as it is presented, because that's a factor most of us will have to face. The big question seems to be the cost-of-money number (the 18 percent). That number is a general requirement for an investor-owned utility which has a responsibility for a return to its investors. Also, to make a return on investment internally requires that costs be covered. For a public utility - municipal or federal - the cost of money is reduced by a factor of almost two. Thus, in a federally sponsored project or a federally run organization, a relatively good COE value appears, and it is quite attractive. The essence of the problem is that if wind turbines can't be made economically satisfactory under this type of examination by the customer, then we may not have a product.

Q. What is the speed variation of the design?

A. It is a constant rpm machine.

Q. Have any market surveys been made to determine how many utilities would buy your machine at the 2.4 cent kilowatt hour rate?

A. We haven't done what might be called an exhaustive market survey. We have been in communication with several utilities who say that if these numbers can be realized, they would be more than happy to buy them. There are indications in the studies that we have done (which admittedly are crude), that there are sufficient wind area sites to make this machine economically attractive. If they could be produced fast enough, we could probably put up around 20,000 machines.

Q. What is the cut-in velocity of your design? One of the largest systems cuts in at wind velocities of 14 miles an hour, while in our program, the smaller machines started at 10 miles an hour. We feel there is a lot of useful wind at that lower level.

A. We start at 10 miles an hour at 30 feet. At the hub height, it is something like 11 or 12 miles an hour. At 50 feet, for the Rocky Flat machines, I think it is around 10 miles an hour or less.

Q. What is the annual production rate that was used in your assumptions?

A. About 200 to 250 units.

Q. Could you explain why a 50 hertz system is specified for your machine?

A. Most areas in the world have 50 hertz systems. In fact, the United States is one of the few countries in the world that runs at 60 cycles. The generator for use in this country would have a few more teeth on the gearbox so that it could operate at 60 hertz.

Q. Are you using a planetary gearbox?

A. Yes. We are currently planning to use a planetary gearbox system, although it isn't absolutely certain. There is some disagreement among our own designers whether to use a parallel shaft or some combination of planetary and parallel shafting. The particular design that is presented here did have a planetary box. It was found to be the cheapest and lightest arrangement.

Q. What is the conversion factor between 30 feet and the centerline?

A. We use the shear gradient that was contained in the MOD-3 proposal request last year.

#### COMMENTS:

On the fixed charge rate, the 18 percent number is roughly representative of utilities which account for 70 percent of all the electricity generated (not capacity) in this country. Furthermore, if a machine can become available at the estimated cost of 2.4 cents per kilowatt hour, if the wind resource in the utilities service area is assured, and if the reliability of the units is also assured, then the utilities will certainly buy these machines.

The wind conversion system engineering field is never static. It is always moving and there are many facets to explore. Much attention and work needs to be done on the indirect contribution to capital cost. Also, it is necessary to verify and pursue the items in the denominator of the COE expression. Attention must be paid to the details right from the start, and total dedication is needed to the equation in order to make these machines happen.



Figure 1. - WTS-3 concept (artist's rendition).

## **WTS-3 SPECIFICATION**

<b>Rotor Diameter</b>	<b>255 Ft. (77.6 M)</b>
<b>Tower Height</b>	<b>262 Ft. (80 M)</b>
<b>Generator</b>	
<b>Type</b>	<b>Synchronous AC</b>
<b>Rating</b>	<b>3.0 MW</b>
<b>Capacity</b>	<b>3,750 KVA</b>
<b>Voltage</b>	<b>6,000 V, 3-Phase</b>
<b>Frequency</b>	<b>50 Hz</b>
<b>Maximum Survivable Windspeed</b>	<b>125 MPH (55 M/S) At Hub</b>
<b>Design Life</b>	<b>30 Years</b>

Figure 2

## **WTS-3 CHARACTERISTICS**

- **Downwind Rotor**
  - **Teetered**
  - **Full Span Control**
  - **Fiberglass Blades**
  - **Infinite Fatigue Life**
- **Free Yaw**
- **Soft Tower**
- **Low Maintenance Requirements**
- **High On-Line Availability**

Figure 3

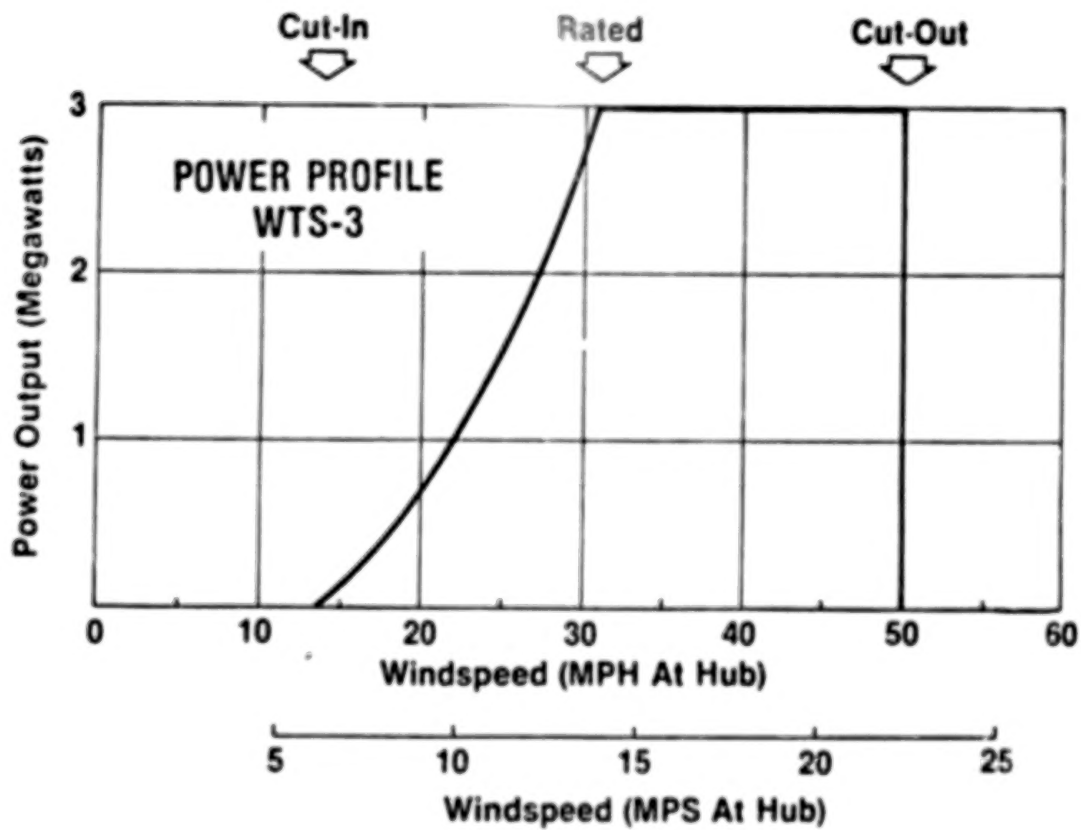


Figure 4

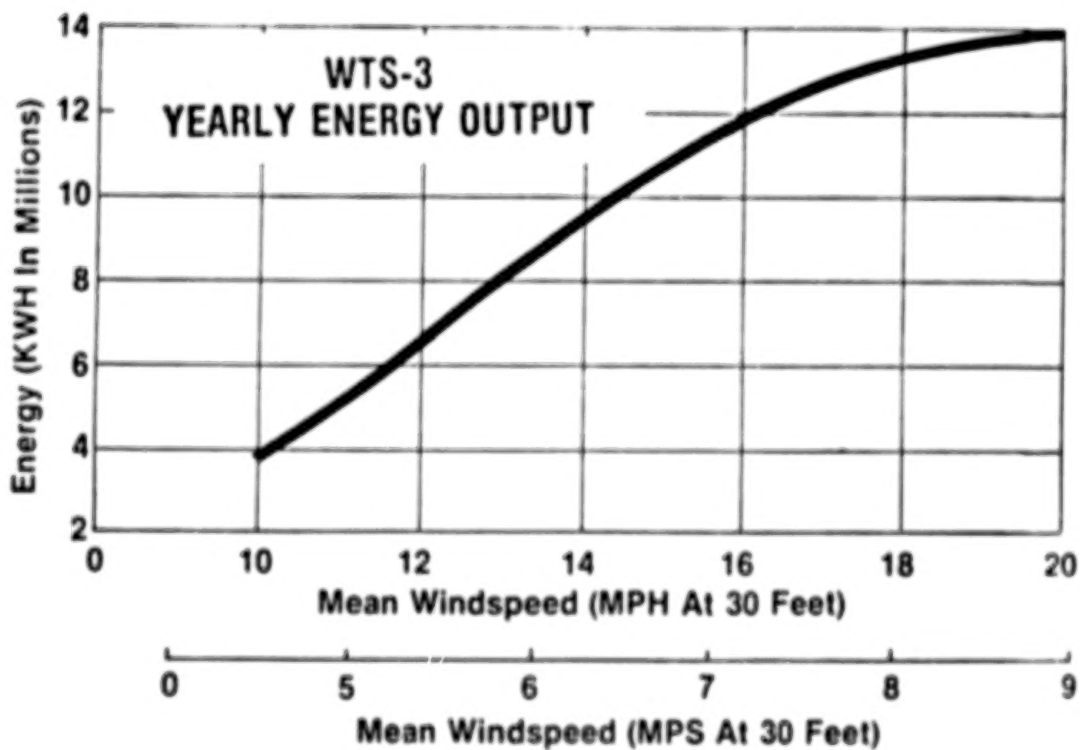


Figure 5

## WORKING GROUP SUMMARIES

### (a) MOD-0 Tests

John Glasgow and Donald Cooksey,  
NASA Lewis Research Center

This working session consisted basically of an overview of the test program and results of the MOD-0 experimental wind turbine at Plum Brook. Both earlier and new data and experiences were presented. Test topics included the steel spar blades, soft tower, fixed pitch rotor, upwind rotor, and electrical system. Future test plans were also outlined.

The discussions stressed what was learned about the performance characteristics of the machine and its relation to predicted or design behavior. Particular note was made of the higher stall point (more torque) observed with the fixed pitch configuration and of the less-than-expected stability with the free yaw and rigid hub configuration.

The discussions raised no controversial points concerning the instrumentation and tests. More testing of the upwind rotors and tip control was suggested.

### (b) Analysis Techniques and Rotor Blades

David A. Spera and James Feddoul,  
NASA Lewis Research Center

The main subject of discussion was the status of the development of specialized computer codes for calculating performance and dynamic loads in wind turbines. Secondly, there was an open discussion about blades and rotors.

As to the status of code development, there is a list of available computer codes which NASA can provide to anybody who is interested. These codes and their capabilities were discussed. Both proprietary and nonproprietary codes are included for horizontal axis wind turbines. It is desirable to keep this list as current as possible. NASA solicits contact from those who have developed computer codes which, with verification, could be used for calculating loads and performance of horizontal axis wind turbines. The LeRC Wind Energy Office regularly informs what technology is available. Furthermore, if any one is interested in performing a service such as calculating loads for people who are designing machines, that also should be communicated to LeRC.

As far as the discussion on rotor blades was concerned, the major question that arose was to this effect: If teetering was considered for the MOD-1 and rejected, why was it accepted for the MOD-2, or why wasn't it accepted for MOD-1? There are a lot of reasons for that, and many of them were programmatic. But, the most important reason is that teetering should not be considered as one isolated item that does or doesn't go into the rotor. Teetering must be



supported by other systems, and as Dick Douglas mentioned, teetering in the MOD-2 rotor was supported by a soft quill shaft, by tip control, by an electro-meric bearing, and by other special hardware in the rotor. Teetering is a package deal. None of these special supporting components was available in the MOD-1 design. That was the main result of the discussion on rotor blades.

### (c) Vertical Axis Wind Turbines

Richard Braasch,  
Sandia Laboratories

The group discussed the vertical axis wind turbine. One of the main questions raised was that of the status of dynamic analysis tools. The basic result is that such techniques are now being worked on, but they do not currently exist. The status of performance analysis tools was also discussed. There was some discussion on the basic structure of the vertical axis program. Of specific interest was the funding level. The primary concern was that, since the activities were scheduled in series, very slow progress will result in this program.

Power qualities were discussed extensively by the utility people in the panel. There was concern about the starting of these machines. It was indicated that neither the horizontal axis machine nor the vertical axis machine is really self-starting. Both need power in substantial quantities to start. It was felt that the vertical machine does require significantly more starting power.

Also, there was some discussion on the use of wind turbines in remote locations, especially in applications where only single phase power is available. In particular, it was questioned whether equipment is really available to support fairly high power single phase wind turbines, that is greater than around 10 kWe.

Other general items discussed were: the nature of such meetings; economic issues; and the need for making comparisons between the different kinds of technologies. The present tendency is to reduce comparisons to just cents per kilowatt hour or dollars per installed kilowatt. Perhaps there should be comparison studies based on fatigue life of vertical and horizontal machines. Longer fatigue life is better, because lifetime is basically built on fatigue. Also, significant differences in operating and maintenance requirement for these two machines should be identified.

The question was raised why the design specifications for vertical axis machines have been different. The answer may have two parts. First, there has been some conservatism in selecting the design numbers. Secondly, there has been more iteration of machine designs in the horizontal axis program, which has resulted in more rapid changes of specifications. Also, some of the requirements came directly from DOE and they weren't changed. However, this is not a severe situation. It can be changed relatively rapidly, and there should not be any problems in reducing conservatism and discussing softer designs.



Some material was presented on reducing conservatism, but we have not thoroughly looked at what a soft design really means.

There was quite a bit of discussion of economic issues regarding what is the optimum size machine. The number appeared to be something larger than 1 megawatt. There was a factor of more than 10 in difference of opinion among the participants, roughly between 1 and 10 megawatts. One of the participants summarized the considerations in the following way. For wind machines, the volume of material basically grows approximately as the cube of the size of the machine, whereas the energy grows with the square of the size. However, a large number of fixed costs tend to make small machines more expensive than large machines. Furthermore, these fixed costs have a decreasing impact on the cost of energy with size. Therefore, eventually the cost must increase, because it is difficult to beat a cube with a square. The size at which that will happen depends a great deal on what is assumed for fixed costs. This is one of the nice little variables that can be used to produce a desired size. We should be careful about that, and be sure the fixed costs are credible.

There was considerable discussion about whether simple is really cheaper, and if it is, how do we quantify it. There should be more to it than just saying it. Many people felt that it is very difficult to quantify "simple."

Another thing that came up is that a discussion item should be placed on the agenda of one of these conferences that deals with the real total data base for the systems that are being tested. We have considerable data of a nature that is not presented at conferences because we believe it is technically too complicated. It might be appropriate for future presentations on these different machines to really cover the entire data base, that is, the set of analysis tools available, and the manipulations performed on the data. This background is quite considerable. As is known, there is much more to it than transforming power measurements into  $C_p$  curves and making strain gage measurements.

#### (d) MOD-OA Operating Experience

Richard K. Shaltens and Arthur Birchenough,  
NASA Lewis Research Center

The session was basically a presentation of operating experience with MOD-OA at Clayton, New Mexico, and the discussion of these experiences. Many of the start-up and operating problems that were incurred during the first year of operation were covered. Many pieces of data were presented with regard to blade loads, icing conditions, and fluctuations in power. The gamut of all the things that were learned at Clayton was covered. Also discussed was the impact of the problems and failures on future designs.

There was considerable discussion and clarification requests for the various items presented. A major question was how to predict mean wind variations for the start-stop cycle (e.g., when will there be sufficient wind to sustain the start). Surprise was also expressed at the extent of the ice built-up on the blades - up to an inch thick. A year-end report covering the first

year of operation at Clayton currently is being worked on. The exact publication date is not known.

#### (e) Lightning Protection

H. Banksitis, NASA Lewis Research Center  
Herbert W. Gewehr, Kaman Aerospace Corporation

The discussion opened with a question and rambled through a gamut of subjects. Basic items covered were direct strikes on the blades or tower, resulting damage, and induced voltages to the structure and electrical systems. Most of the time was spent discussing direct strikes on composite blades.

The opinion projected was that if a composite blade does not have electrical wires or any kind of a conductor inside of it, there should not be a problem. However, the dielectric property of the composite could cause charge focusing and result in structural damage. No one knew for sure what the damage would be. Hopefully, tests of a piece of blade should determine that one way or the other. It was recommended that the tests be done.

In general, the group's opinion was that bearings should be protected, although the exact methodology of protection was not discussed in detail. Various schemes were mentioned.

It also appears that induced voltage effects on electronics and electrical equipment should be investigated. How this is to be done was not discussed. Induced effects were of concern because of the use of microprocessors in either controlling the machines or recording the data. Microprocessors are affected by indirect or induced voltages from either a strike directly on the machine or near the machine. There is some literature that might be available, and a good literature search might be a proper start.

A point was raised concerning design practice. Composite blades supposedly have a benefit as far as TV interference is concerned. However, the installation of lightning protective system might ruin that benefit by causing TV interference that shouldn't be there. This characteristic should be taken into account early in the design and, if possible, not compromised.

Feedback is needed on actual strike experience. This relates to the old aircraft days when pilots compared notes as to how many times and where they got zapped. Such communication did boil down to an experience data bank. A handbook was written and the information became available to the people facing a lightning phenomenon environment. Thus, there is a need for feedback of information between ourselves as far as wind turbines are concerned - specifically, how many strikes have been experienced and what are the immediate assessments of damage. If the structure is taken apart for maintenance or for whatever reason, what kind of inspection ought to be carried out to determine the damage that was caused by lightning to bearings or to something that is not readily visible from the surface. How this is to be done was not really determined; it was just discussed.

An opinion was expressed that composite blades with tip controls would obviously need protection because of the conductive members inside of the spar. Metal blades should be carefully handled, and proper electrical bonding should be provided as a part of the protection scheme, especially in the joint areas. A cost benefit analysis of the scheme should be determined.

A general opinion was that this area of interest should be an ongoing activity at workshops. Participation of people interested in input to such areas as lightning protection, strike experience, and damage assessment would be very beneficial.

#### (f) Drive Train Components

Richard L. Puthoff, NASA Lewis Research Center  
Richard R. Douglas, Boeing Engineering and Construction Company  
Allen P. Spaulding, Jr., WTG Energy Systems, Inc.

Three types of gearbox configuration were discussed: the parallel shaft-box that has been used on the MOD-0 and MOD-1 machines; the planetary type gearbox that was selected for MOD-2; and the integral parallel bed type design proposed for the MOD-1A design and used on smaller machines by WTG Energy Systems. The advantages and disadvantages of each were discussed. Basically, there was one conclusion that the planetary gearbox does result in saving overall symmetry. It was also decided that that doesn't necessarily make it the right choice for a particular machine. Gearbox selection still has to be an individual function for every machine.

Specifications for the gearbox were discussed. The need for a definite load and service spectrum to be provided by the gear manufacturers was stressed, so that longer service life could be achieved.

Other components of the drive train were then discussed, with particular emphasis on those that impact the gearbox. Included were generators, universals, three-blade rotors, and larger gearboxes. Soft and stiff quill shafts were discussed. It was noted that the nature of the gearboxes is such that as the machines tend to slow down, the torque increases faster than the energy rate. This should be taken into account, particularly on large machines where speed is compromised to reduce gearbox size.

## SYSTEM DEVELOPMENT STATUS DISCUSSIONS

### Vertical Axis Wind Turbine Panel

Panelists: Robert Thresher, Oregon State University  
Emil G. Kadlec, Sandia Laboratories  
Daniel K. Ai, Alcoa Laboratories  
Richard Braasch, Sandia Laboratories  
Robert O. Nellums, Sandia Laboratories  
R. J. Templin, National Research Council of Canada

Question. - Have any studies been done on TV and microwave interference with vertical axis machines?

Comments. - A study was done on the vertical axis machine by the University of Michigan (Dr. Tom Senior). Scale model rotors of both vertical axis and horizontal axis machines were built and tested for interference effects, primarily to television. The entire television spectrum was covered. The results indicated that all of these machines exhibited interference effects. There was no configuration that eliminated the problem. However, there is a significant difference in the interference effects of the different machines. The interference effects at any given frequency for a vertical axis machine are approximately one-third that for a horizontal axis machine of comparable size. The TV interference effect is on the synchronization and the electronics and not on just the amplitudes. The report of this study is about to be published.

The situation is basically that most of the TV signals that are broadcast today are broadcast as horizontally polarized signals. When the blades on a horizontal axis machine are in the horizontal position, they reflect back to the TV set antenna. Because of the blade configuration in a vertical axis machine, there is less effect from this phenomenon. These results are still preliminary information from experiments that were run on subscale models. Much more data is needed in this area, and a lot of data has to be taken on actual machines.

Question. - The horizontal wind systems discussed are in the megawatt range, and the vertical systems are generally in the hundred kilowatt range. Is that because the vertical system is lagging in development compared to the horizontal system, or is it that the vertical system will never compete with the horizontal system at large output values? Is there some limitation due to cost or technical reasons?

Comments. - The Canadian National Research Council has built the largest vertical axis wind turbine so far. It is a rebuilt version at a rating of about 230 kilowatts. This machine constitutes a large jump for the designers, because the biggest size they had run before was on the order of 10 kilowatts. Although the accident with the earlier machine was very embarrassing, it is stressed that the accident had nothing to do with the question of large scale machines.

As to what the optimum size is, it appears that a pretty flat level is involved, at least for cost, somewhere in the megawatt range. However, it is still too indefinite to determine accurately. The size question is basically a matter of dollars. The difference in funding in these programs is about the same as the difference in size. However, it does not appear to be because there are not good proposals for large systems.

Design studies for megawatt VAWT machines are being conducted in Sweden, since it is felt that now is the right time to start. It is believed in that country that much information is available, and there is a strong desire to get the hands dirty on testing a large size machine, especially the dynamic loads and gust sensitivity.

Question. - Why is there no MOD-2 type vertical system in the planning stage?

Comments. - An increase in the size of the Darrieus machines is expected in the program. However, the general trend is analogous to the way the horizontal axis machine program started. A start is made with small ones. Then, as one gets comfortable with these machines and with the computer models that analyze them, the move is made to a larger machine. We are certainly more comfortable with the megawatt machines now than a year ago.

Question. - Could a multimegawatt machine be built right now? Are all of the needed information and tools ready to go?

Comments. - There was some discussion in the working group about that point. Basically, if events are put in series, accomplishment will take a long time. For example, consider the history of the wind energy projects in general as a starting point. All of the tools did not exist to properly quantify everything for the propeller machines when the large sizes were constructed. It was mainly a parallel effort, and it produced a running machine. There has certainly been discussion as to whether the first things built were right. But, that is what evolution is all about. Some risks are necessary to get something done.

The general preference is to basically continue that same approach with the vertical axis machine. Even though all of the necessary tools are not available, it is important to proceed anyway. That is, do something a little bit risky, and develop some things in parallel. That way, some answers will be obtained, but it probably won't be a perfect machine. Such an approach is preferable to placing everything in series and never producing anything. Basically right now progress is in series with the vertical axis machine.

An industrial design and cost study was done last year in Canada. That study did not uncover any technical reasons for not considering the megawatt range. There are aeroelastic and other problems, but at least as long as one is willing to pay the price for adequate design, it can be done.

The estimated cost of a prototype in the low megawatt range, for design, construction and tests in Canada is something like \$15 million stretched over 2 or 3 years. That's more money than the budget can cover. The annual budget



for energy R & D in Canada has been about a million dollars a year for the last couple of years. The megawatt prototype is a big step, and it still hasn't been approved.

Question. - The vertical axis machine would appear to be one option for the collection of energy and its conversion to electricity that is not convenient for the horizontal axis machine. Specifically, is it possible with a multiplicity of these machines in a field, to pump water in a loop at the bottom of the machines, and somewhere in the loop have a conventional water turbine, the control of which is routine by now?

Comments. - In general, the primary system cost is not the generator, but there is significant cost in the transmission. Thus, in this proposed arrangement, the electrical generator is centralized at some slight savings in cost, while the power is converted in a relatively low efficiency hydraulic system. In contrast, power conversion efficiencies for presently-used mechanical transmissions are 70 to 80 percent. The hydraulic concept might be possible with a large gear ratio, something like several hundred to one, in a large megawatt system.

It is not known whether any studies have been made of such a hydraulic system. It may have some potential. There is, however, something similar. The Department of Agriculture under DOE funding has installed some wind systems that are mechanically coupled to pumps through an electrical motor. The wind turbine drives one end of the electric motor which drives the pump at the other end. As the wind rises, it unloads the motor, and when the wind dies out the motor picks up the pump load. Such designs are being experimented at one of the U.S. Department of Agriculture's facilities at Amarillo. So there has been some activity on mechanical coupling to wind turbines, but not exactly the type in question. Further information can be obtained from Lou Liljidalh at the U.S. Department of Agriculture in Amarillo.

In another related effort, Sandia Laboratories has a subcontract with Valley Pump Company which involves the capability to either generate electricity or attach the turbine directly to a pump located on the ground. This is a very definite advantage for the vertical axis turbine.

Question. - Why hasn't the Giromill turbine concept been considered for large scale systems?

Comment. - Two small Giromill turbines are currently being developed through contracts with Rockwell at Rocky Flats. If these machines are successful as determined by DOE/WSB, then the larger machines may be attempted.

Question. - Is Reynolds number scaled in the model tests?

Comment. - The Reynolds number is very important in model work. Unfortunately, wind tunnel model tests can't usually duplicate full-scale Reynolds numbers. As a practical measure, one uses the largest models and tunnels available.

Question. - As far as blade cleanliness is concerned, is it known how much effect spoilers and other such things have on blade efficiency?

Comment. - There isn't much of a feeling for that effect because the spoilers are different on each turbine. There haven't been any good measurements to date. The Canadian program has a few small turbines with and without spoilers that can provide a direct effect. It is believed that in most cases the effect was small, an overall effective blade drag coefficient of around 0.002. The effect is small, but not negligible.

Question. - This question is about the load carrying structure of the guy wires. It is desirable to eliminate these guy wires. Is any work being done on this? Can we have towers like the Giromill tower? Such consideration is especially important for offshore sites. It would be rather difficult to have a hundred meters of guy wires outside the unit in offshore applications.

Comment. - As far as is known, the joint venture of Alcoa at Clarkson College is probably the only vertical axis wind turbine of its type that is free-standing. It has a double shaft with a stationary inner steel tube and a rotational outer aluminum tube. By comparison with the current Alcoa design, that structure is way overdesigned because it wasn't known what kind of problems would be encountered. There was considerable concern about the bending moments.

At Alcoa we agree with the opinion that we ought to try further, because the limitations are not known. We have gained confidence in operating the Clarkson machine, and we don't think that the stationary tube needs to extend all the way to the top. Freestanding certainly has an advantage if the turbine is to be mounted on top of a tall building. In fact, many calls have been received lately inquiring whether turbines can be installed on top of apartment buildings and the like.

Question. - Who is rebuilding the Magdalen Island machine, and if there is a change, what are the reasons?

Comment. - That machine is being rebuilt by DAF-Indal to essentially the same drawings, without change of the basic structure. Many of the same parts, which tend to be expensive, such as the small parts, bolts, steel cones in the tower are being salvaged. Changes in the design that are being made include the method of latching the spoilers and the move of the brake from the high speed shaft to the low speed shaft.

Question. - In the case of the extruded blades, are they extruded straight and then bent?

Comment. - Yes, but not on a stretch press. The blade is stretched to straighten the extrusion. A three-point bender is then used to form it into an approximate troposkien. As far as we know, the section remains symmetrical after bending. Very tight tolerances are kept in those blades. That was a concern in the very beginning. All kinds of ways of filling in the hollow voids before forming were considered, but were unnecessary.

The blades, consisting of four extrusions, were delivered to Albuquerque from Alcoa. Then Alcoa came to Albuquerque and bent them, taking about two-and-a-half hours per blade. There are some residual stresses left in the curved section when it is bent. This is one of the reasons why care should be exercised concerning the desired shape the blade is to be bent to. It is important not to leave residual stresses in the blade at the root, which is the high stress area.

As far as the history of blade bending is concerned, the first blade that was bent by Alcoa was the Kaman Blade. The D-spar was bent first and then the 6-inch chord blade, a section of which is on display. This blade has a very tight radius. It is probably the worst situation they had. They started out very cautiously with ten pieces of extrusions in the hope that three good ones would be obtained. It took one engineer and two technicians a full 8 hours to bend one blade. That's for a 6-inch blade, so real progress has been made in the past 3 years.

There is another experience with blade shape. A 1-meter wind tunnel model was built with blades made of spruce. They were initially straight and then steamed and bent. It was found, however, that after bending, particularly on the thicker ones, the section was no longer symmetrical. Since then, a system was developed to make the blades of plastic and cast them in a mold which is of the proper shape. Beautiful blades are now produced, but they are not nearly as strong.

Question. - Instead of being concerned about a symmetrical section, why should not camber be desired when, in essence, the curvature of the flow is being changed?

Comment. - For wind tunnel testing, it is important to know what the blade shape is rather than have a random profile. With regards to flow curvature, a set of blades is being built on the Sandia 5-meter machine which corrects for this flow curvature. Another set will be built and then a decision will be made about cambering. That should be something reported on in the near future.

Question. - The DAF machine worked about over 200 hours before it blew down. Was the failure connected with the utility, and were there any voltage fluctuation (flicker) problems or any problems associated with the interconnect?

Comments. - The number of hours accumulated on the Magdalen Island turbine at the time of its collapse was just 238. There are now two 50-kilowatt machines, like half-scale models, running. They each have a 1000 hours now, mostly since about December. The voltage ripple is pretty low and, of course, the predominant frequency is twice per rev, which is about 1 hertz. That is about the worst from the power company's torque point of view.

The voltage ripple is a small fraction of 1 percent. The shaft power ripple, of course, is larger, and it's not a constant percentage of mean power. It is more like 20 percent of maximum power. There were no complaints of any kind from the power plants.



Question. - The same question about voltage ripple is asked of the Sandia machines. Did the dynamics in the two- or three-blade vertical axis machine cause excessive flicker, and how many hours were logged on the Sandia 17-meter machine?

Comment. - The Sandia machine was operated something like 500 hours. No flickers were observed, and there were no visual effects of torque isolation. Extensive measurements were made on the voltage oscillations - they were on the order of 0.2 to 0.4 of a percent oscillation. This value falls within at least our technology of a half percent limitation, which is an acceptable voltage fluctuation.

Question. - What are some of the structural and dynamic problems that might be encountered with a large cantilevered vertical axis machine?

Comments. - To investigate this question, a start would be to talk to the wind turbine people at NASA. They must have the largest data base on cantilevered wind turbine structures. However, it is believed that the major advantage of the vertical axis machine is the fact that both the blades and the tower are connected at both ends. One opinion is that there isn't any reason for giving up those advantages. The cables are not felt to be that serious of a problem, and there should be an overwhelming reason for eliminating them. The interest in getting rid of cables for offshore mounting is understandable. However, there is some feeling that dealing with the offshore installations at this time is questionable.

It may be more important to first build the wind turbines that work well on the ground, and then consider floating them in the water. It is recognized that some countries have inadequate amounts of unoccupied ground in desirable siting areas. That is not a problem in the United States, but it may well be a serious problem in countries like Sweden. In such instances, it would be encouraging to see the results that would be obtained. It is a major advantage to hold the top of the tower, and at least in the United States, that approach should be continued wherever possible.

Question. - Are there any limits to cables? When will cable sag become a problem?

Comments. - There are cable problems, and there are bending problems associated with the vertical axis machine. We find if we want to have cantilevered machines, the tube size may be a problem.

As far as vibration is concerned, experience with the Alcoa machine pertained to how rigid the tower is for holding. The first tower was mounted upside down, and because of that it was very soft. There was concern it would topple and shake the silo loose. Since then the tower was righted and made more rigid. That eliminated the vibration problems, and it worked out fine. However, occasionally we do call Clarkson College to see if it is still there, and it still is. Cables may become questionable for sizes above 300 feet. In the case of the guy cable, there is a problem of applying sufficient tension in the cables so it will control the cable droop, which is produced by gravity. In

using the steel cables above a 300-foot diameter system, it's difficult to eliminate the droop and to obtain the desired stiffness at the top of the machine. This can be circumvented by going to a cantilevered system, or by not building systems that are large, or by accepting the droop. The best approach is not clear. The droop condition is somewhat arbitrary; that is, the droop shouldn't be greater than such and such. But, proper values haven't really been established.

There probably is a limit to the use of guy cables. That limit is believed to be in the range of 300 foot diameter machines or greater. For the sizes that are being considered today, there should be no technical limits to the use of a cable system.

There are other issues involved. Cables have been a problem in the past, and many people would like to replace them with something that may, in effect, be worse. The basic problem with the cantilevered system is the rotating tower with a massive bending movement at the bottom. Remember that a wind turbine is basically a fatigue testing machine. When a rotating tower is pushed sideways and all of the load is taken out at one end, a substantial problem exists. It doesn't mean it is impossible, but the design stresses in the tower must be very low.

The other problem is the bearings needed and the separation between the bearings at the bottom in order to keep the bearing size reasonable. If both bearings are kept together, they will be huge. If they are separated, then deep foundations are needed. These are all eliminated when a tiedown system is used. That is a very inexpensive way to hold down the tower. Another point is that maintenance people like the idea that in the vertical axis machine, all of the machinery is near the ground. With guy wires, there is a high compression load on the top, as well as on the lower bearing. The bearings might thus run red hot on occasion, which would present a problem. In that event, it is desirable to have all the machinery low where it is readily accessible.

Question. - Has any consideration been given to the use of composite blades on vertical axis turbines?

Comments. - There is much interest at Sandia in materials other than aluminum, not because of a dislike of aluminum, but to explore the potential for cost reduction in the future. It is true that there was a considerable reduction in cost between the first set of blades for the Sandia 17-meter machine and the extruded aluminum blades. However, in considering this comparison, it should be noted that the earlier blades were built from a completely different set of design requirements. A part of their high cost was due to the design requirements and not to fabrication. Alcoa believes that when composites can match aluminum costs, the aluminum prices will be lowered.

In reply to an earlier question, Alcoa has a two-fold interest in wind turbines. One is to sell mill products, and an extrusion is a mill product. They have been in that business for many years and intend to continue. The second reason is the search for new ventures. Recently Alcoa's chairman of the board and president approved a new venture committee to look into the possibility of diversification. If diversification happens, then whatever blade serves

the purpose best will be used. At this point, for company interest in vertical axis machines, aluminum is believed to be a good way to go. Composite blades should also be very well-suited for use for a variety of reasons. However, care should be exercised in comparing first cost composites with blade costs after a considerable amount of process development. One other consideration: after 30 years in service, aluminum blades can be sold as scrap for some return of cost.

#### Horizontal Axis Wind Turbine Panel

Panelists: Ronald L. Thomas, NASA Lewis Research Center  
Robert Thresher, Oregon State University  
David A. Spera, NASA Lewis Research Center  
Richard H. Poor, General Electric Co.  
Richard R. Douglas, Boeing Engineering and Construction Company  
B. Maribo Pederson, Technical University of Denmark  
Sven Hugosson, National Swedish Board for Energy Source Development  
Hans A. Thiele, M. A. N. New Technology

Question. - We have now reached the point where several two-bladed and three-bladed machines have been built and operated. We have heard reports from people who have both types. Could we have a discussion on two-versus-three blades, and, in particular, the reasons for the selection in each design?

Comments. - As far as the MOD-1 machine is concerned, the NASA specifications required two blades because MOD-1 is a scaled-up version of the MOD-0. However, we did look at three blades in several studies and found that in large diameter systems, no benefits are derived from three blades. One of the working sessions concluded that perhaps small diameter machines might benefit from three blades. However, we have not at G.E. been able to find any cases where three blades are cost-effective for a 200-foot system.

For the MOD-2 machine, two blades were not prescribed. In fact, a very complete two-versus-three-blade study was made at Boeing. Three blades appeared attractive because of the reduced torque. Most of the system cyclic loads are eliminated with a three-blade system. Unfortunately, for large systems, the blade cost becomes a very dominant part of the total cost.

In the case of MOD-2 approximately 10 percent would be added to the system cost for a third blade for a gain in energy of only 1/2 percent. Solutions for the dynamic problems were obtained, but it simply turned out to be the additional cost of electricity that negated the three blade system. According to the studies, the rotor weight increases at a faster rate than the cube of the rotor diameter. There is a cube-square sort of rule working here, and blade costs become proportionately greater. That is why small machines can consider a third blade and take advantage of benefits.

It was also found in the studies at Boeing, that not only did the third blade cost too much, but in order to optimize power, a three-blade machine must be run much slower. In terms of performance, the slower speed for the same power means higher torques. Thus, the gearbox, which also has this same square-cube relation, increases its size appreciably, and that is the second most costly component in a large wind turbine. As a consequence, both blade cost and gearbox cost increase with a three-bladed rotor.

In Denmark, three bladed machines are partly a tradition. More significantly, the Danish machine is a relatively small machine, and the cost analysis indicated only something like a 1 percent increase for the third blade. However, if a megawatt size machine were to be built, the rotor would be changed to a two blade design.

When one considers the efficiency that can be gained by adding a third blade, one is tempted to look at it very seriously. However, in view of the dynamic changes that have been introduced, such as the beauty of teetering, one questions how these advantages can be obtained with a three-blader. The answer is they can't. The hub becomes too cluttered, and the three-blader cannot be compared favorably with the advantages of a teetering two-blader. Interestingly enough, the attractiveness of the three-blader on a rigid system is not only that of preserved simplicity, but that it avoids the dynamics problems of the two-blader. Those dynamics problems also disappear when a teetered two-blade configuration is adopted.

In summary, many three-bladed machines have been built in the past, and they have simpler dynamics if the machine has a rigid hub. However, if a fairly large machine is to be built, the three-blade configuration is not the way to go. The panel's conclusion is that two blades are preferable if the dynamics are understood. With a softer system, such as a teetering hub, the advantages of two blades and a lower cost system can be obtained.

Question. - The MOD-0A's and the MOD-1's have full pitch control, the MOD-2's are being built with 35 percent tip control, and the WTC machine is really a fixed pitch machine that relies on blade stalling for power control. These approaches were first discussed back in 1972 and 1973. A lot of effort was exerted to determine the right approach, and here in 1979 we are building all three kinds of machines. One would think the question would be resolved by now, but apparently there are proponents of all three ways. What are the advantages and disadvantages of the methods of pitch control?

Comments. - For the MOD-1 machine, in retrospect, it appears that full span pitch control is more costly. The pitch mechanism system is elaborate on the MOD-1. As a result of most recent work, a change in the pitch control mechanism was made to make it smaller and hence less costly. It also enabled the use of a teetered hub. If a teetered hub is attempted with total span control, the mechanism becomes very complex and expensive. That is the reason why G.E. did not elect to use a teetered hub on MOD-1.



Full pitch control is useful in large machines designed for lowest solidities in the blades. Such machines become difficult to start, and full blade pitch control makes it easier to start. Full control also provides good response for gusts.

For the MOD-2, teetering was selected before tip control. The teetering scheme, the type of old barn door hinge arrangement that is seen on the model, was defined before the conversion to tip control. Tip control fits that particular hub better than it fits the MOD-1 type of hub. A 20 percent tip was considered at first, but the machine couldn't be started and stopped adequately. The final design was a 30 percent tip in order to provide the machine with some reasonable startup time and some reasonable control in an emergency. It was found that tip control does aggravate alternating torques (2/rev). However, partial tip control is a more economical system than full span control.

It is of interest to note the lengths and weights of the pitchable airfoil for some of the different machines. The MOD-0A, with full blade pitch, pitches 58 feet of airfoil that weighs 2300 pounds. On the MOD-2, with partial span pitch, 45 feet of airfoil that weight 10,000 pounds is moveable. The Smith-Putnam wind turbine controlled pitch on three quarters of its blade length which was 67 feet long and weighed about 16,000 pounds. Thus, it is the size of the item that is being pitched that is important. It is not just full or partial pitch, which are only relative terms.

Once the hurdles of structural integrity are overcome, the really important matter is control. How is the machine started, and how is it stopped? The runaway rotor is the number one safety concern. Stopping a fixed pitch rotor is a real problem. That is why every fixed pitch rotor has some stopping device on it, like tip flaps or a parachute. Thus, it is felt that control is really important, and that is what determines the length of the airfoil that is pitched. It is not a matter of full span versus partial span control; it is how much is needed to start and stop the rotor.

Question. - Is there any difference in gust response for the two pitch control approaches?

Comment. - The amount of teetering that will be obtained with a small pitchable section will be much higher than if there is full blade pitch. This is what was referred to in an earlier comment about the large cycling torques on the low speed shaft which arise from teetering. A fixed pitch rotor or one that has three-quarters of it fixed in pitch, will teeter to about twice the angle for a given gust or wind shear as a fully pitched rotor. This presents problems for clearances and for the low speed shaft design.

Question. - Are there any comments about fixed pitch?

Comment. - We looked very hard at fixed pitch machines at Lewis and found that one disadvantage is that the generator and gearbox must be sized to take the maximum energy extracted from the rotor before it starts to stall. Sizing the generator and gearbox for that situation does not necessarily produce the optimum size generator or gearbox with regard to lowest COE. It was

felt that the wrong size gearbox and generator would result. One of the advantages of pitch control is that the power can be limited, which results in a better size gearbox and generator. This is one of the considerations that might exist for the Darrius machine, which has no pitch control.

Question. - Is it possible that the blades might continue to rotate after they are stalled? Some aerodynamic shapes, the cylinder may be one, will continue to rotate in the free-stream after the critical rotational speed is reached - a sort of autorotation. Might large rotors also do this?

Comments. - If the question is concerned about the possibility of the rotor running away and destructing when the load is lost, that is not a concern. Fixed pitch machines have some device such as spoilers to unload the blades. Also, if rotational speed increases, angle of attack is reduced which moves the operation further away from stall.

With pitch control, the blades would be feathered, that is they would be turned with the chord essentially parallel to the wind. There are certain winds that could cause the rotor to rotate sporadically, and gusts could turn the rotor one way and back. However, continuous rotation is not aerodynamically possible under these circumstances.

Question. - Another point that comes up frequently is the question of upwind versus downwind rotor location. There are machines in the program that have both upwind and downwind rotors, and it would be interesting to discuss some of the reasons why.

Comments. - For the German Growian machine, the specifications required a downwind configuration. There wasn't time to study the question to make a selection. It was mostly a matter of past history with the machine. The Danish machines both have upwind rotors. No good reasons were found to place it downwind.

The MOD-1A study looked at upwind and downwind teetering rotors, and there was not too much variation in loads. The comparison was very close, and it could have gone either way.

A downwind configuration was proposed for the MOD-2 machine based pretty much on past history and ignorance. As a more careful study was made, it was found, as suspected, that the loads weren't that different for the upward or downward location, particularly with a teetered hub configuration. Since about  $2\frac{1}{2}$  percent more energy was gained because of the absence of wind blockage from the tower, and since no load differences were found, there was no reason to go downwind. Thus, the upwind configuration was chosen.

Question. - Do utilities qualify for additional investment tax credit of 10 percent for special energy properties as defined in the 1978 Energy Act?

Comment. - A preliminary check with DOE people and also with personnel from EPRI indicated that this does not apply to utilities. The provision was

aimed primarily at homeowners, and there was a \$2,000 limit in the legislation that was passed. Any additional information is welcome.

Question. - The MOD-1 is rated at 2 mW at 24 miles an hour, while the 300-foot diameter MOD-2 requires 26.4 miles per hour. Why the difference?

Comment. - A basic problem exists concerning wind speeds at 30 feet and at hub height. When talking about a specific machine, it is convenient to talk about hub height (or the rotor centerline). When talking about sites, the 30 feet reference is convenient. The difference is that the MOD-2 reaches rated power of 2500 kW at 26.4 miles an hour at the hub centerline, while the MOD-1 numbers were given at 30 feet. MOD-1 reached rated power of 2000 kW at 32.6 miles per hour at the hub.

Question. - In the MOD-2 presentation, the factors in calculating wind power cost estimates were carefully specified. These costs must be compared with competing power sources. What are DOE's and Boeing's estimates for the costs of competing sources, and what rates of general inflation and fuel cost escalations were assumed in calculating these costs over a 30-year lifetime?

Comment. - One can't speak for DOE, and it is not sure that one can speak for the Boeing Company either. This is a fairly complex question, and there are too many costs involved to be challenged at this session. In the Boeing calculations for cost of energy, the government specified the formula to use, which includes capital costs, fixed charge rate, O & M, and annual energy. Some levelizing cost estimates were included. It should be appreciated that a big part of the cost of wind power is the original procurement costs, and a comparatively small part is operations and maintenance. This latter part, which is an escalating factor, is not as important as it is in other systems which involve costs for fuel which escalates strongly. That is really all that can be said on the subject at the moment. DOE has made some studies and Boeing has also, but the data are not available here.

Wind machines costs will have to be in the range of 2 to 3 cents a kW-hour to make a significant energy impact. It is believed the MOD-2 will come close to these goals, and the next round of machines should get well into that range.

Question. - In regard to the question on costs, it sounds like the 2- to 3-cents per kW hour level came down as a specification from DOE. There appears to be considerable controversy as to whether wind power needs to get that cheap to be competitive. What are the inflation rates that were assumed in the cost of fuel for competing power sources, and what assumptions were made in terms of inflation or escalation of that price above general inflation? It appears that this must be precise in order to establish the 2- or 3-cents per kW hour. Are we saying that the Boeing machine can get to that range?

Comment. - The 2- to 3-cents value is the goal established by DOE. For the MOD-2 machine, it was required that the machine had to show a cost less than 4 cents a kW hour for 100 units with a moderate production of several per month. The 4 cents a kW hour is based on the capital cost of the machine times

a fixed charge rate of 0.18. That is the number specified by DOE, and is also the fixed charge rate for all emerging technologies. That includes taxes, interest, the cost of the machine, and so forth. That, plus O & M are then divided by the annual energy to give the cost in cents per kW hour based on a 30-year life machine.

Question. - Large machines are understood to be designed for utility sector applications, but we generally talk about the cost and performance of a single machine. Shouldn't we talk about many units, like on a wind farm, and what the cost will be, say for 200 or 500 units? Isn't that what the utility sector is looking for?

Comment. - The value of less than 4 cents a kW hour for the MOD-2 machine assumes that there are 25 machines interconnected in a cluster and tied into a utility network. A single unit MOD-2 machine would cost a lot more because of the operations and maintenance cost. With 25 units, a full crew is available 24 hours a day to service the machines. Also, the cost of operations, material, and spares can be shared.

It will certainly be necessary to look at larger numbers of machines. Such results are just not available now. The study that produced the 25-unit MOD-2 cluster at less than 4 cents provides the best number available. It is not certain that increasing the number of machines to a hundred or a thousand will really decrease the cost by an appreciable amount. Most of the benefits appear to be realized with numbers like 25.



## BLADE DEVELOPMENT STATUS DISCUSSIONS

### Vertical Axis Panel

Panelists: Richard Braasch, Sandia Laboratories  
David A. Spera, NASA Lewis Research Center  
Daniel K. Ai, Alcoa Laboratories  
Emil G. Kadlec, Sandia Laboratories  
William N. Sullivan, Sandia Laboratories  
Robert Thresher, Oregon State University

Question. - There was mention of a blade flutter problem with one of the Sandia test models. Could this be elaborated?

Comments. - About 3 years ago at Sandia, a set of wind tunnel models was built. One of those models, which was 2 meters in diameter, was used to test aerodynamic performance. Plywood blades were constructed for this particular wind tunnel test series. Plywood was used partly because it was thought to be an interesting technology, and it was a convenient means for providing different chords and airfoil shapes. When the plywood blades were tested in one configuration with a relatively low solidity, there was a fairly violent flutter instability that occurred in the rotor at around 250 rpm. Since the normal operating speed for this rotor is in the range of 500 to 600 rpm, this flutter certainly precluded its usefulness as a testing device.

The aluminum blade did not exhibit the same characteristics. This observation led to a lot of R & D on the subject of flutter. It included contacting Norman Ham at MIT to do some analysis to determine why it happened and how it could be avoided. It is believed the NRC in Canada also initiated some tests on small models based on Sandia's unfortunate wind tunnel test results.

The overall conclusion reached by Norman Ham was that the problem was caused by the very low torsional stiffness of the plywood blades. Also, the problem should not occur in the aluminum blades because they have a substantially higher twisting stiffness. Actually, the tests at NRC in Canada on small wind tunnel models found no problems with the aluminum blades. For the time being, the use of wood blades was stopped, and all future designs will involve aluminum extrusions.

These studies suggest that for the configurations being investigated at Sandia, flutter would not be a major problem. For example, the 17-meter machine has never exhibited flutter. Also, the Magdalen Island machine, operating at roughly twice design rpm during its accident, never experienced flutter. Nonetheless, more should be known about this phenomenon, particularly as future designs tend to explore less conventional configurations.

The 2-meter wind tunnel model was recently tested again in the field at Sandia to try to get a better handle on the flutter problem. The plywood blades that were made for the wind tunnel test were installed on the machine in the field, and sure enough they fluttered at about the same rpm as in the

wind tunnel test. The tower was then stiffened, and we have since been unable to make the plywood blades flutter. It was somewhat surprising that tower torsional stiffness would have such a pronounced effect.

The test turbine that fluttered is continuously variable in speed from zero to something greater than 600 rpm. In looking with high speed photography at the conditions that did flutter, it was not obvious at all that there was any tower twisting. However, when the tower was changed the flutter did disappear. Thus, the tower twisting must be very small to cause it, and apparently it doesn't take much.

As far as the plywood blades are concerned, we should be capable of controlling the flutter by at least the aforementioned means, and probably by others. In the future, particularly if alternate blade technologies besides aluminum extrusions are considered, and if the solidity of these machines is changed, the flutter could become significant. It is important that this phenomenon be watched, and that improved capabilities be developed to cope with it.

There is some further comment on the flutter analysis that was performed for the 17-meter machine. Alcoa provided Norman Ham with vibrational modal analysis which was done in two ways. The first was to consider the blade attached to a rigid frame. The flutter critical speed was then calculated. As was shown in the Alcoa presentation, nominal operating speed is  $51\frac{1}{2}$  rpm, and we would like to be able to increase that to about 60 or 61 rpm (the allowable turbine rpm is 75). Based on Ham's calculations, the flutter critical speed is 124 rpm. So, it is quite a bit higher than the range the machine is expected to operate in.

Secondly, we considered the entire system as elastic, that is, capable of making elastic response. The tube, the cable, the drivetrain, were all included in the system. Based on this system, the vibrational modes were calculated again as far as the 23rd mode. It turned out that flutter is influenced by the 21st mode or something near it. The flutter critical speed was lowered from 124 to 108 rpm for the elastic system, so we are still quite safe. There will be a report out on this analysis.

Flutter may not be unique to wood blades. A number of other blades have been made to flutter recently at Sandia. However, in general, it was found that flutter speed was substantially higher for aluminum blades on a rotor with the same solidity. Norman Ham's conclusion was that this difference is due to the higher ratio of the shear modulus to the elastic modulus of aluminum relative to plywood.

It should be pointed out that, as was mentioned earlier, wood has a property that it shares with fiberglass. For homogeneous materials like steel, the shear modulus is 40 percent of the tensile modulus. In composites, of which wood is one, the shear modulus is one or two orders of magnitude different than the tensile modulus. Thus, in the case of a plywood blade, it might be desirable to orient the layers to achieve the desired properties and thereby control flutter. For example, just placing some of the layers at  $\pm 45^\circ$  might increase the shear stiffness sufficiently to suppress the flutter.

Question. - What are the constraints on extruding aluminum blades? For example, is there some constraint on the web thickness as compared to the length or the size of the cells in the blade or the number of cells? What really controls that process?

Comments. - The first controlling factor is the cylinder size, that is, the cylinder than can be fit into the press. Secondly, it is the pressure required. For example, using the Alcoa 14,000 ton press for the 29-inch blade, a 25-inch cylinder is used, which gives a pressure of 56 ksi. That much pressure is needed to push the blade through. There would not be sufficient pressure if a larger cylinder was used. A 29-inch blade can be obtained from a 25-inch cylinder because a special die makes the metal flow. The length of the extrusion is controlled by the weight of the billet. Usually it's around 5000 pounds for the 14,000 ton press.

The other controlling factor is the material. For the so-called porthole or bridge dies, the 6063 alloy is preferred. 6061 Alloys can be used, but not the 2000 series. A 2000 series billet would have to have a solid section rather than a hollow section. Wall thickness is also a factor for the different materials. For the 6063 material it's probably a quarter of an inch wall. That kind of thickness is typical.

There is a window from about 0.20 inch to maybe 0.35 inch. The exact range is rather complicated to pin down, since it depends on the exact shape and size, etc. But, there definitely is a min-max range. If the web thickness is too large or one side thick and the other thin, then extruding becomes difficult. It is desirable to have a balanced configuration so that the mass flow of the metal can be controlled. For example, a blade thickness of a half inch is probably outside the window of the 14,000 ton press. Wall thickness from 1/8 of an inch to about 0.08 inch could be done on a smaller press. It all depends on the exact shape and size blade desired.

Question. - What is the cost of the 29-inch blade in terms of dollars per pound?

Comment. - That blade weighs about 20 pounds per foot. The cost today is probably around \$1.75 a pound extruded. The die cost is around \$20,000. These prices are as low as they will be. Alcoa's policy is that once the die is made, any length can be extruded. Should there be any damage to the die, it will be replaced free-of-charge. Thus, the die is purchased only once regardless of wear on damage.

Question. - In order to mainize the risk of cost redesign and such things, it is essential that before going into production, a well-disciplined stress analysis and experimental program be conducted. The consideration of such factors as retention areas, joints, critical stress sections, and full scale blades means large size test facilities of the kind that probably do not exist anywhere today. Isn't it time for NASA to start thinking about what components should be submitted for this kind of testing, and what kind of test programs are reasonable?

Comments. - Component and full-scale testing for vertical axis blades is certainly one of the items that will be mentioned later under R & D requirements. As far as Sandia Labs is concerned, there are no plans in their program to do such testing within the available funding. Also, NASA has no plans for this type of in-plant fatigue testing of full-scale blades prior to going into the field under actual conditions. However, that question is certainly a good starting point for the R & D discussion.

Question. - With respect to large cost-effective vertical axis machines, how important is the use of constant rpm in order to avoid dangerous resonances?

Comments. - The main motivation for running at constant rpm is to provide a regulation to the drivetrain power. In effect, the utility grid provides the automatic control system for the machine. The desire to avoid crossing of system resonant frequencies that would occur in a variable rpm operation was also considered. That is another good reason for going to constant rpm. It hasn't been proved nor is it believed that a variable rpm Darrieus rotor could not be built because of dynamic problems. But it is an added risk and complicating factor. Quite a bit of work has been done at Sandia to try to conceptually examine just exactly what kind of benefits might be obtained from a perfect control system that operates a variable rpm rotor. The basic conclusion is that the benefits being considered are not that great. It is on the order of a 10 to 15 percent improvement in energy collection with the most optimistic assumptions that can be made. The question of how such a system can actually be controlled has not been addressed. The hardware problems are probably immense, particularly on a Darrieus rotor without aerodynamic controls. The basic method to produce variable rpm operation on a machine without aerodynamic control must be through load control. In this case, massive torques are involved, which need to be switched on and off. This is easy to do on paper, but for the hardware that is required, it is a rather disturbing prospect.

Question. - When large machines are considered, very long slender blades are involved. If no strut supports are present, there is a problem with high stresses and fatigue at the blade base. Is there some way of increasing the blade size in that area to reduce the chordwise stresses in the rotor at the root fittings?

Comments. - An obvious answer to this issue is the use of the tapered blade. There is a drift in that direction, but very cautiously. In Emil Kadlec's presentation, there was talk about tailoring the blade wall thickness, particularly if multiple sections of blade were involved. Such an approach should contribute to helping that problem, in that the blade can be tailored to the load with a potential reduction in stresses and costs. Then it can be asked if wall thickness can be varied, why not the chord. Varying the chord doesn't seem nearly as attractive because of the problems of joining sections with stepwise changes in chord.

The idea of a continuous taper would require a rethinking of manufacturing technology. It could be in the future that improved manufacturing technology will appear so that taper would not impose a serious compromise. However, there is no question that the vertical axis program is oriented toward

the simplest solutions for the moment. The main reason there is reluctance to jump into tapered sections is the desire to prove that it can be done with untapered sections first.

Actually, in the maximum stress area, the trailing section of the blade can be changed while the forward part of the section is maintained. This would help the fabrication. This may well be a possible approach. In fact, there is actually a little bit of tailoring on the Kaman blade on the 17-meter machine.

Question. - Has any thought been given to a welded steel blade? The reason for the question is that the cost of aluminum extrusions is not a trivial one.

Comments. - Yes, they have been considered. Straight blade sections for the Sandia 5-meter machine were actually made. However, there were a number of problems with them. A curved section couldn't be made with the steel welding. The price is right for the raw material, and it might be possible. But, there doesn't seem to be any serious proposals at this time.

Question. - Have buckling tests on the extruded aluminum blades been conducted, and if so, are the data available?

Comment. - Buckling tests have not been conducted on the aluminum extruded blades. The NRC in Canada has conducted buckling tests, but not using extruded blades. Beam buckling when the blade is formed was looked into at Sandia. One of their senior metallurgists as a consultant recommended introducing webs, and that seems to alleviate the problem.

Question. - What is the most probable failure mode for the extruded aluminum blades? Which mode should receive the most attention and the most careful analysis?

Comment. - Vibratory stress and fatigue in the blade at the trailing edge where the blade joins the tower in normal operation is felt to be the most severe condition and the one designers are probably most uncomfortable with. Another place would be the root area where the blade is connected to the tube.

Question. - Is there a fundamental reason why intermediate supports can't be used on the guy cables?

Comment. - That point is being worked on at Sandia. At the present time, there is no certainty that any known support will work. The Canadians have tried a number of things, and they have not been too successful. However, it may still be a possibility. Also, some sort of dampening mechanism might be appropriate.



### Horizontal Axis Panel

Panelists: David A. Spera, NASA Lewis Research Center  
Bradford S. Linscott, NASA Lewis Research Center  
Robert E. Donham, Lockheed Aircraft Service Company  
Timothy L. Sullivan, NASA Lewis Research Center  
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Meade Gougeon, Gougeon Brothers, Inc.  
Herbert W. Gewehr, Kaman Aerospace  
John Van Bronkhorst, Boeing Engineering and Construction Company  
Gordon Davison, Boeing Engineering and Construction Company

Question. - In regard to the MOD-2 blade, what capacity brake press was used to form the root sections? Is this process a normal commercial one?

Comments. - The press capacity is about 1500 tons. The width of the blade on the press is the most significant factor for MOD-2. At Boeing we are looking for 25-foot presses to accommodate the largest rotor plates.

Relative to the second question, the brake press operators are used to making simple shapes. They have made half sections for telephone poles and light poles with small tapers. Their method is to first lay out straight bend lines on the steel plate and then incrementally work the plate through the brake using the lines as bends. It takes just a few hours to work each side of the plate and a crane to handle the material. The process is commercially very feasible. We had three vendors do that test section that was shown earlier. But the main point is it is not usually the press tonnage that is critical. If the brake is wide enough to accept the part, it will probably have capacity to form parts even up to 1 inch thick.

Question. - Why was A633 with low sulphur selected, as opposed to A516, GR70, or some other more easily available material with lower lead times?

Comments. - The main selection criteria for the MOD-2 rotor steel were static strength, fabricability, fatigue strength, and costs. The final candidates were all above 36,000 psi in yield strength, had a low carbon content for good weldability, together with bend diameter-to-thickness ratios of 2 or less, and had reasonable Charpy impact values at low temperatures with ductility greater than 19 percent.

A 633 Grade A steel was chosen because in addition to the above characteristics, we could specify a fine-grain structure with low sulphur and low copper. The latter requirements enhanced cleanliness and aided in the prevention of crevice corrosion respectively. Both of these are, of course, important in fatigue resistance considerations.

A 516 material was not included in the original screening of 18 A-type steels, so its relative technical desirability was not examined. However, acceptable quotes were obtained on mill delivery times for the A633 plate.

Question. - In reference to the 150-foot blade, does Kaman have a reliable nondestructible method of detecting the type of laminate flaw which caused the failure of the MOD-2 size blade?

Comment. - The laminate flaw was clearly visible on the outside surface of the spar, although its significance was not realized prior to testing. It was found that X-ray techniques will reveal fiber orientations and can be used to determine the extent and significance of visible imperfections of the type that caused the crippling failure.

The 150-foot spar was built by Structural Composites Industries, Inc. There was a so-called defect or "bulge" in the spar wall at Station 45 (wrinkled fibers), where it is felt that buckling failure on the blade initiated. That type of defect, in the opinion of S.C.I., was encouraged by the choice of winding pattern used to wind the 150-foot spar. It was an involved sequence of events that led to that choice of winding pattern, but it may not have been a good winding pattern for the structure. The resulting composite was unbalanced. The pitch of the winding was always righthanded instead of alternate layers of right and left hand pitch which are ordinarily used. That pattern was chosen for good reasons, but it had the effect of encouraging this "bulge" with successive plies of parallel windings and by not allowing it to be flattened by the usual alternating plies at right and left hand pitch angles. Therefore, the "bulge" is not viewed as a defect as much as a consequence of a less-than-optimum winding pattern.

Question. - How many men and man hours are required to assemble one of those 40-foot steel blades, from the time all of the parts are available until they are ready to be shipped? Also, how much was learned between the first and second blades that were made?

Comments. - There are essentially three separate parts to the spar section in that blade: a root spar; a main spar; and a tip spar. All of the ribs are the same, and there are about 30 of them per blade. The skins are all put on in one complete section. In other words, the leading edge is brake formed and then draped over the entire blade and riveted on. The first blade took about 2 weeks to make and employed about 10 people. The second one took about a week and employed about two people. That gives an idea of what kind of learning curve can occur in a small company.

Question. - For the MOD-1 blades, it was noticed that both a composite and a steel blade were manufactured. Might there be some basis for drawing a comparison of the weights, costs, and production of these blades?

Comments. - Both the MOD-1 steel blade with its fabricated trailing edge and the MOD-1 composite blade have the same weight, approximately 20,000 pounds. It was difficult for both of those construction methods to meet that 20,000 pounds restriction, but they did. As far as costs are concerned, a second pair of MOD-1 fiberglass blades is estimated to cost about \$400,000, which is \$10 a pound. This is a rough estimate based on Kaman's experience with the 150-foot blade. The cost of the MOD-1 steel blade is really not comparable because the manufacturing process was a special one used only because of the severe weight restriction. For example, one whole side of each blade spar was machined to

reduce the wall thickness between weld lands. It's estimated that another pair of blades made this way would cost almost two million dollars or about \$50 a pound. Obviously, no one is planning to make another pair of blades this way.

Question. - Concerning the Kaman blade, is only a visual nondestructive evaluation being used on that blade?

Comment. - In addition to visual inspection, witness samples are cut from the ends of the spar for gravimetric and mechanical properties tests, and Barcol hardness tests are performed on the spar surface in other areas. Tap tests and conventional ultrasonic tests are not very effective on thick laminates, but ultrasonic immersion tests can be used effectively to detect voids and delaminations with the laminate. X-rays, as mentioned previously, can also be used to detect voids and fiber orientations. Tap tests, witness samples, and through-transmission of light were used on the afterbody panels to check adhesive bond lines and honeycomb core fillets. Conventional ultrasonic techniques (Fokker tests) were used in suspect areas.

Question. - Even in the most sophisticated blade there are defects that need to be detected one way or another. A critical flaw in composites is a subject of considerable controversy. What are your criteria for an acceptable flaw?

Comment. - The blade was designed to utilize the properties and quality of commercial processes, not only in the spar, but throughout the blade. In view of the occurrence of the local bulges in the spar, and the resulting crippling failure, an additional one-third knockdown factor will be imposed on the allowable stresses to be used for design of spar areas critical for buckling. Thus, the MOD-1 spar will be designed to withstand an ultimate bending moment equal to two-and-a-quarter times the design limit load which, for the 150-foot blade, was produced by 165 miles-an-hour hurricane winds at the hub.

Question. - It has been observed by a number of people that there is significant modulus degradation in glass epoxy. Was any modulus degradation detected in the fatigue tests that were run on the TFT samples?

Comments. - No modulus degradation was detected in either the small specimen tests or the quarter-scale root and specimen tests. Applied strains in the laminate and applied load throughout both series of tests were monitored, and no degradation prior to failure in the small specimen tests was detected. There were no failures and no modulus degradation in the quarter-scale tests.

In further regard to modulus degradation of composites under cyclic loading, this effect can probably be used as a quality measurement method to detect when a composite blade is reaching the end of its useful life. If the modulus is reduced, then the blade will become less stiff and we should be able to detect this change in stiffness. This would provide a good means of knowing when a blade needs to be replaced.



Question. - Were any creep studies done for the Kaman blade?

Comment. - Yes. Creep specimens were put in an environmental chamber for 1000 hours at 95° and 90 percent RH, and no measurable creep deflection was found. The stress level was 4000 psi, corresponding to the static droop stress on the blade while parked in a hot, wet environment.

Question. - Can more detail be given on the type of turbulence that was discussed in regard to the MOD-2 blade? What kind of data were used to evaluate the blade loadings for that type of turbulence?

Comments. - Type I loading was one-per-rev cyclic or alternating loading, and Type II was the gusts. The hub section of the MOD-2 rotor has a teeter mechanism which attenuates the high frequency turbulence. The tip control attenuates the low frequency loading from the rotor.

The various types of cycles that are being described here are newly coined words, so that everything that is being discussed is really in a state of development. The type II fatigue cycle is one in which the wind has been blowing at one level, and then a gust comes through the machine. The gust completely engulfs the machine, raising the wind speed to some other level at a rate which is relatively slow in comparison with the rotor speed, and then subsides again. The Type II stress cycle would extend from the lowest stress point before the gust hit to the highest stress point during the occurrence of the gust, and then back again to the low stress point. That would be one Type II cycle.

The way that cycle is evaluated for stresses is to take the time history of the gust and put it through a computer code that determines how fast the controls will react in alleviating the thrust and torque loads associated with that gust. The peak thrust and peak torque are determined, and loads are calculated as if they were a steady-state condition. As a steady state condition, the peak thrust, the peak torque, and a compatible wind velocity are taken. There will be some high load point for this condition. The low load point is taken from the cycle before the gust, and that's the model of a Type II gust response.

There is another aspect to this question, namely the frequency of the gusts and what model is used to determine how many gusts there will be in the 30-year lifetime of the machine. There is a probability density for the occurrence of gusts of various sizes. That model has to be determined and used by the analyst. A model for that does exist.

Question. - What might be the upper limit of blade length that blade technology could produce? That is, what is the largest machine that could be constructed with the different technologies, and what would be the inherent limiting factor?

Comments. - There doesn't seem to be any practical reason why composite or any blades couldn't be made larger than 150 feet. Other factors probably get into the picture; such as transportation, handling, erection on the tower, and

considerations not associated with structural limitations on the blade themselves.

As far as aluminum blades are concerned, there are detailed designs of a 200-foot rotor which were actually offered for sale by Lockheed. The manufacturing process was an early consideration prior to these offerings. There is no discomfort in stating that 200-foot diameter rotors can readily be made. It is also believed that a 150-foot rotor blade could also be made. What might happen is that changes in the movable outer span approach will take place which will limit span requirements. The advantages of the hub detail design will increase span options. Thus, it is expected that the rotor diameters can reach virtually any size, as the Boeing Company has indicated. It might very well be that the need for a blade with a radius larger than 100 feet or so will never really occur. However, it is certain that the required rotors can be made in aluminum as well.

The problem of utilizing wood in very large sizes was examined at Gougeon, but not to the point of doing detailed design studies. The biggest problem which arises from scaling to larger sizes is that the gravity-induced loads increase more rapidly than the structural strength. Thus, the question of the largest achievable size will be limited ultimately by the specific properties of the material being used (i.e., strength and stiffness per unit mass). That holds, of course, if various other practical considerations do not limit the growth first.

With respect to wood specifically, it has good specific properties. It is also interesting to note that the largest wooden structure of this type that's ever been built in the Hughes airplane, which has a wingspan of about 320 feet. The examination of the 150-foot blade, and what would have to be done to scale the design work to that size, indicates that wood should be practical for that size range.

Question. - What is the longest length utility pole that is made today?

Comment. - The largest utility pole made today is 250 feet long with six laminates at the base. In contrast, for the roll-flow process that was described earlier, the longest length that has been made is 40 feet. Thus, to obtain longer lengths with this process, it will be necessary to weld two pieces together. The technology could probably be advanced to make longer spars using the roll-flow process.

The WTC rotor now is 80 feet in diameter. Preliminary studies have been done on 180 to 200 foot sizes, with no discomfort in that range. Steel rotors can certainly be made much larger than that. It is not known what the upper limit would be, but it certainly could be as high as any other material.

Regarding size limitations of composite structures, SCI has built rocket motor cases up to  $22\frac{1}{2}$  feet in diameter. If that were the chord of a blade at the root (approximately double the chord of the 150-ft blade), it would extrapolate to a 300-foot blade length. For very large turbines, the blades will be fabricated on site. For a large wind farm with tens or hundreds of large

turbines, there is no reason why a portable plant could not be set up to manufacture the blades at the site. That would eliminate the shipping problems that were mentioned earlier.

Question. - Could the different blade technologies provide an indication of their estimates of the eventual cost levels for horizontal axis blades in terms of dollars per pound?

Comments. - Boeing would be comfortable with something less than \$5 a pound for their type of welded steel blade, although that is a long way from the numbers that were heard earlier. It may be closer to \$3 a pound for a hundred blades. It is interesting to note that water towers are built for a dollar a pound erected on the site.

For the Kaman 150-foot blade, the actual cost of building that first specimen was \$10.33 a pound. That is the recurring cost of that blade, excluding tooling, design, development, and all of the other nonrecurring factors. That cost was projected to an average over the next ten blades at just under \$7 a pound, and for the hundredth blade at about \$3 a pound.

Although cost projection for a thousand blades could not be made by Lockheed, it was noted that a nonrecurring cost at less than \$25 a pound was offered for a delivered second set of blades 200-feet in diameter weighing a total of about 35,000 pounds.

Gougeon projected a unit cost between \$5 and \$7 a pound for 100-foot wooden blades. Longer blades will be somewhat lower than that, probably closer to the \$5 per pound price. It appears that wood has the potential to look better in comparison to other materials at small production levels. That might be the more interesting aspect of wood at this stage.

The cost of NASA's second utility pole blade was about \$15 a pound. The projection for limited production is that this could be reduced to about \$10 a pound. For large production quantities, the cost might be reduced to \$3 to \$4 a pound, with an appropriate investment in tooling.

WTG Energy Systems has steel blades available at \$3 a pound. As was pointed out yesterday, it is certain that the price can be reduced to  $1\frac{1}{2}$  a pound.

Question. - The MOD-2 blade was shown earlier with some zero margins on high cycle fatigue. In the Boeing paper, a statistical impossibility was quoted for areas critically stressed and having zero defects. How can conservatism and unlimited life philosophy be square with what was heard yesterday?

Comments. - The margins shown were for 200 million cycles, and were based on flaw growth models using fracture mechanics techniques. It is assumed that the material could be cleared for all flaws smaller than the ones that were used in the fatigue test specimens. The spindle also has a positive margin of safety and has no built-in defects. The bearing likewise has a good margin.

Question. - The overwhelming impression from the detailed examination and design of the MOD-1 blade is that the selection of materials and the detailed inspection processes involved were not consistent with what would be called low-cost technology. Is that impression correct? In the event it is correct, what is seen in the future for these types of inspections? Can they be eliminated, or made cheaper, or what?

Comments. - The observation is correct that there was a great deal of inspection made on that blade that may not be involved in a production run. That was a first-of-a-kind situation that MOD-2 production will not have to contend with. There was also more complexity in the MOD-1 blade than the MOD-2 blade due to the combination of both metallic and nonmetallic structures. It is known that two people can read the radiography records and come to two different conclusions. That is especially true with the ultrasonic method. It would not be feasible to do on MOD-2 production units that type and extent of inspection that was used on MOD-1. With respect to the nonmetallic part of MOD-1, there was a lot of inspection of bonding that had to be contended with. As was mentioned before, it would be better to have good process control all the way through.

Basically, the MOD-1 blade is a one-of-a-kind blade that was made under some very severe weight and stiffness restraints. It was very expensive and difficult to build. It has no relation to the MOD-2 program. The MOD-2 is a very economical blade to build. It has very little inspection, and works at much lower stress levels. Thus, we are talking about two different blade programs, one with severe stiffness and weight restraints, and the other with almost no weight restraints.

Question. - When comparing materials which differ in density by a factor of 10 to 1 as in the case of wood to steel, it is very misleading to talk about price in dollars per pounds unless it is assumed that every blade of a given length will have the same weight. To be fair, the price per blade should take into account the different densities and different weights of the materials. This leads to the general question, what is the benefit of lightweight blades in the system? Is lighter better?

Comments. - Lightweight, in itself, is not better. Lower cost is what is needed, especially a lower cost of electricity. The blades should be designed with the COE requirements for the complete wind turbine system. If it can be obtained with a heavier blade, fine. If a soft tower is designed, a heavier blade might be needed - as long as low cost techniques are used.

The approach at WTG Energy Systems is to use heavy blades, probably the heaviest for the size rotor than anybody else. That was found to be cost-effective, and it was also better from a running standpoint. It is easier to control the machine during wind gusts with high inertia in the rotor. We are very happy with a heavy rotor.

The blade weight question depends on the configuration of the particular machine. A general statement cannot be made whether lightweight is good or bad. In certain instances, there are some benefits to the system of going to a lighter blade, such as reducing total rotor weight and tower weight, etc.

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For instance, in the MOD-0 and MOD-1 machines, which have rigid hubs and two blades, a torsional frequency problem might arise if the rotor inertia gets too high. A fairly high torsional stiffness would then be necessary in the tower. In that instance, a lighter blade would save weight in the tower. However, once a teeter hinge is used, that problem vanishes.

If a constant rpm machine is used, not very much benefit will be obtained with a heavy rotor from the standpoint of the energy storage in the flywheel. The speed has to be changed to remove stored energy from a flywheel. There probably are some advantages to a lighter blade, all other things being equal. A lighter blade has less material, but obviously, the economics has to be the bottom line. A heavier blade will impose higher loads on the control systems, for example, on the pitching mechanism or on the tip control mechanism. Moments can be generated with a heavy blade which may be more of a problem than moments with a lighter blade. That may impact system cost in addition to basic blade cost. Another point about a heavy blade is that there will be some effect on the low speed drivetrain that will cause some increase in the size of the driveshaft and the bearing supports.

In going to colossal sizes, as was mentioned previously, the one load that does not scale uniformly and that gets worse with size is the gravitational load, that is, the bending moment due to gravity. If the machine size is doubled, it is as if the force of gravity was doubled. The blade can only be built so big before it becomes overcome by its own weight. Thus, for very large blades, weight is intimately tied to the limit of blade size. However, at this time we cannot adequately say where this limit is.

There are several other questions that are rather unanswerable at this time. One of them is the effect of polar inertia which requires that kinetic energy be stored in the rotor. That loss in energy during starting and stopping for each start-stop cycle must be paid for out of the energy losses in lost operating time. Also, the authority of the rotor aerodynamics as a function of rpm compared with rotor inertia when teetering systems are used, could result in stop banging, which might be a difficulty. Furthermore, as far as the yawing system impact is concerned, the tower design stiffness is far less important. The interface stiffness between the nacelle and the top of the tower in the case of the MOD-0 dominates the yawing frequency determination.

For the initial installation on MOD-0, the two blades weighed a total of 4000 pounds, while the nacelle system weight was 28,000 pounds. This suggests that we might look for other places to save weight rather than the rotor. In the final analysis, the cost of energy is the ultimate criterion which determines what is better. The blade design must be consistent with the requirements for the complete wind turbine system.

R & D REQUIREMENTS  
FOR  
VERTICAL AXIS WIND TURBINES

Comments by Richard Braasch  
Sandia Laboratories

At Sandia, we have put together a brief outline of what we think the R&D needs are for vertical axis machines. Under technology development, there is the need for improved structural dynamic models. We feel there is a need also for improved aerodynamic performance and load modelling.

In the area of fatigue life assessment, there is need for experimental data to be combined with the structural dynamic calculations that are being performed. This is necessary in order to validate these tools while progressing to larger machines. We think there is a rather strong need for a method of integrating all elements to obtain a correct life prediction assessment for these machines, rather than just looking at maximum stresses and how many cycles might have occurred. That is, stress behavior should be accumulated correctly and weighted correctly for determining fatigue life.

One of the significant advantages of vertical axis machines (Darrieus type) is that the left hand portion of the power coefficient curve just so happens to result in controlling the peak power out of the machine. We feel there is a need to learn how to tailor this portion of the curve to fully exploit that advantage. For example, we would like the parameter  $\max K_p$ , which relates to the maximum power point, to occur at a slightly higher tip speed ratio. This would reduce the peak power of the system. We think there are some interesting things to do which will have an impact on the overall system design.

Also, we believe there is a need for hardware fabrication experience, particularly with large blades, since there is still a desire to reduce the costs of blades. The blades are quite inexpensive for relatively small machines, but when we look at the size of the blades required for large machines, we are not sure that they too will be inexpensive.

The torque tubes that are involved in the vertical axis machine will have relatively large diameters for large machines. We think there is some need to understand how to optimize these tubes in terms of wall thickness versus position on the tube. There is more to learn about manufacturing and handling tubes of this size.

There has been some concern about being able to ship and handle the tubes because of the relatively thin walls. That leads to the question of making a truss tower rather than a tube tower. The studies that we've done show that a truss requires a larger diameter than a tube, which means the only way a shipping advantage can be obtained is to basically build them at the site. That comparison really has not been thoroughly studied from a cost standpoint, that is, which one of these approaches would have the lowest cost and lowest weight. We think this is an area that requires attention.

The transmission area has been a problem for everyone. We are readily able to specify transmission torque ratings that don't exist as stock hardware. The vertical axis machine has a particularly bad problem in this respect because of the low rotor speeds and high power peaks. That is one of the reasons why we want to trim the peak power of the machine. This reduces the torque requirements on the transmission and essentially directly reduces the cost, since torque and cost are almost directly proportional.

There are a lot of other things that we would like to do with transmissions that transmission people don't like. For example, we would like to load the entire machine on top of the transmission and have them build the required bearings right into the transmission. In this way, a lot of feel alinement and things of that nature won't have to be done. It's remarkable how tough they say these transmissions are, but they don't want you to touch them. There seems to be some inconsistencies in the way they are built and the way they are used. We need to negotiate more with the transmission people. For an item as important as transmissions, a really competitive environment is needed.

We would also like to point out that often when the cost of energy is assessed, it is very easy to think that things can be built in the field relatively inexpensively. This has been mentioned by many people. However, according to staff members whose job it is to assess erection costs and putting plants together in the field, such activities are startlingly expensive. It is easy to make the cost of energy a lot cheaper if these on-site erection cost estimates are made low. We should be paying a lot more attention to field assemblies, because we think they represent a large cost. Some really experienced people need to look at the erection procedures and be involved in the design. We intend to try to do that in the future.

Then we get to machine development, and we believe that for the vertical axis machine, attention needs to be paid to reducing the conservatism in the design. We have been asked to put some effort in that area by DOE. Also, we think it is a good idea to have a test bed or a machine that can be used for making modifications, trying new ideas, gathering data, etc. and which has a lot of freedom to modify and change things. We currently use our 17-meter machine for such purposes, and intend to continue doing that.

We would like to get busy on a large machine and offer the propeller machine technology more competition in the large size area. We don't have any plans for building a large machine in the near future. Mr. Ancona of DOE talked about two more advanced procurements of the propeller machine, and we are not even talking about procurement of a large vertical axis machine. So, it looks like we will be two more generations behind, and I am not sure a lot of competition can be offered if we get any further behind. We think there is a real need to build one of a credible size.

Comments by R. J. Templin  
National Research Council of Canada

Dick Braasch gave a comprehensive listing of the things that Sandia feels is needed, so I will restrict my remarks to the aerodynamics of vertical axis machines. The first comment is that I do believe in the COE formula and all the terms in it, so my remarks will be in that context.

I believe that the aerodynamic understanding of what goes on in the vertical axis turbine is still at the rudimentary stage. More R & D should be done to improve that lack, not just for the sake of obtaining more knowledge, but because it could be used to improve the energy output of these turbines and thus increase the denominator in the COE formula. As an example of our ignorance or uncertainty, we often don't know the best airfoils to use. We are still using simple, ancient symmetrical NACA airfoils.

The aerodynamics for predicting the effects of major geometric variables such as rotor solidity on overall performance aren't really too bad. However, in those theories, the induced velocity field is only approximately calculated. For example, the momentum stream tube theories do not allow for the difference in induced velocity at the upwind and downwind surfaces of the rotor, and therefore do not take into account the fact that the airfoils are at different angles of attack at the two locations. So, if one inserts available aerodynamic data for cambered airfoils into the theory, one likely gets the wrong answer on the effects of camber. More work, both theoretical and experimental, is needed to develop an adequate flow field for theoretical modelling. The summary feeling is that the aerodynamics of these machines may still be in the infancy stage.

Comments by Daniel K. Ai  
Alcoa Laboratories

Since I represent the private sector, and not government-sponsored laboratories, as in the case of the previous two speakers, I want to bring out some different aspects. In order to press my point, let me tell you about two incidents involving two laboratories. The first one is from Gene Strickland. He was at our lab 18 months ago to assist with the design of blades. He told me about the time he was experimenting with a Darrieus rotor with one of his students. All of a sudden they noticed the load was off. Looking out the window, they found that the rotor was gone. That was how they lost their machine. Then, earlier this year I met a professor from Michigan State University who told me that they have already lost two sets of blades.

What I am trying to stress is that at this stage the Darrieus rotor could be a killer, very much like the airplane in its early days. These rotors are high speed machines, so the blades could fly apart like arrows. Therefore, at Alcoa, we are concerned about the safe operation of these machines, and we keep safety very much on our minds. Alcoa is currently going through a very extensive campaign on safety for two reasons. First, we cannot afford to have accidents. Accidents can cause loss of life and property. The second reason is that Alcoa is obsessed with the thought of being sued. We worry a lot about

product liability. Maybe this factor presents only an indirect impact on the designs and R & D of the future, but we feel that system safety cannot be over-emphasized.

This, then, is my first point. I don't know whether this should be a government-sponsored program or somebody else's responsibility, but I think when we go out to sell turbines, and when we have lawyers preparing warranties, we should be able to tell our clients what to expect.

The other point is the marketing problem. As private companies enter the wind energy business, a large amount of money and time will be devoted to marketing. At this time we still have trouble describing to people what the Darrieus rotor looks like. Private companies may find the job too large for their resources. Thus, a program is needed very much like the solar program, where a large scaled demonstration can be conducted to let the potential users know what the machine can do for them. In the meantime, such a program could help manufacturers to understand a wider variety of applications. Since we have spent so much money and time developing the machines, wider applications would provide more return on the investment. We would like to see a sophisticated well-conceived plan led by DOE to shorten the acceptance time for the benefit of both the users and the manufacturers. I believe such a demonstration plan could have a favorable indirect effect on cost.

Comments by William N. Sullivan  
Sandia Laboratories

I would like to elaborate on the discussion Dick Braasch gave on the machines for the future. The point he was trying to make was the idea of testing a machine for an extended period of time, like the MOD-OA. There is a significant need to get some Darrieus machines out in the field, running and accumulating thousands of hours so that we can gain experiences like NASA has had with the MOD-OA. Such problems are the best education that can be obtained within a reasonably short period of time.

#### Discussion

Question. - It was mentioned earlier that Strickland was working on some advanced aerodynamic theories. To what extent will the items that Jack Templin listed be satisfied by this work?

Comments. - Some of the questions that Jack raised will be at least looked at. This study is a vortex theory, so it will include upwind and downwind rotor effects, which I think is a major issue. However, it is a highly speculative sort of research at this point in time, because it is not clear that the three-dimensional problem will not be so complex as to make the results of limited value for a design situation. As far as improved accuracy over the single or multiple stream tube models is concerned, it remains to be seen if the new approach will be much more accurate.



The problem is being done both in 2-D and 3-D, and of course the 2-D problems are a lot easier. It involves a straight blade of infinite length in which the velocities induced by vortices are tracked everywhere. He does have a 3-D version which is similar in principle, but much more complex.

Question. - Is there any experience with the tangential tolerance, toe in or toe out, or whatever it is called, as far as an actual blade is concerned? If toe in or toe out actually occurred, either inadvertently or intentionally what were the effects?

Comments. - At one time, a wind tunnel test was done at NRC in Canada on a 12-foot diameter model. It was found that one blade was inadvertently toed out about  $3^\circ$  at the equator. After the first test, the blade was remounted to correct the error, and the test was repeated. No difference in the  $C_p$  curve was seen to within the measurement accuracy. However, no systematic tests were conducted on this effect.

There has been no experience at Sandia with toe in or toe out, per se, but some measurements were made of the actual profile before the blades were installed. Fortunately, the blade profiles were so close to nominal that no conclusions could be drawn.

Question. - Today, virtually all the work in this country on vertical axis machines has been done by Sandia, which probably was an excellent choice, considering that with limited funding it is best to keep it in one place. However, in the future, if this concept is to be viable, the procurement or the actual manufacture will be made by the private sector. When do you see any procurements from potential builders of vertical axis machines who would get involved with the development of the entire system, rather than just manufacturing the blades for some of the demonstration models?

Comment. - There is a procurement out right now for the low cost 17-meter machine. The design phase for this machine was completed by a contractor, namely Alcoa. We are currently under negotiations for the fabrication phase. Future procurements highly depend on the budget. We expect that procurement action on a larger machine will start in about a year.

R & D REQUIREMENTS  
FOR  
HORIZONTAL AXIS WIND TURBINES

Comments by Allen P. Spaulding, Jr.  
WTG Energy Systems, Inc.

We have quite a bit of experience with these machines and, at this point, we are not concerned with more hardware development. However, the one area that we feel is going to become increasingly important is the control of line frequency for multiple units in applications involving wind generator penetrations in excess of 50 percent of the installed grid capacity. The particular application we are referring to is the small diesel-powered utility system. These systems in the U.S. have a cost of energy of about 10¢ per kWh, which makes many of them prime candidates for utilizing wind powered generating systems.

Our experience on Cuttyhunk Island has been with a wind turbine that achieves 100 percent penetration levels 25 percent of its operating time annually. We feel that we have an adequate, cost-effective control system for a single wind turbine installation for this type of application. However, we foresee additional control system problems associated with multiple wind generator installations for applications indicating high penetration levels. The DOE machines discussed at this workshop, both installed and under construction, do not, in our opinion, have electric power frequency control systems that are adequate for high penetration levels. Therefore, we would like to see more work on speed control systems for high penetration levels using multiple wind turbine installations.

Another situation specifically related to high penetration levels of wind generators in diesel powered grids, are the thermal and mechanical problems associated with diesel powered generators running for long periods under reduced or no-load conditions. The problems encountered, which relate specifically to the diesel engine, are governor reaction time and "wet stacking." Governor reaction time is the time it takes the diesel's governor to bring the power level from no load to full load. This operating condition is frequently encountered when the wind turbine is operating in a gusting type of wind. "Wet stacking" is literally the pumping of unburned fuel into the diesel engine's exhaust system. This condition is encountered when the diesel engine is allowed to operate for long periods under significantly reduced load. Both of these operating conditions are well understood by both engine manufacturers and utility operators. The traditional technique for dealing with these problems has been to dispatch the combination of generators to match the demand. This technique, however, becomes inappropriate with the introduction of a wind system capable of high levels of penetration.

We have discussed these operating conditions with Detroit-Allison and Delco Division of General Motors. They feel that these problems can be solved. At present the market potential for wind-diesel systems is, in their opinion, not significant enough to warrant the necessary in-house R & D for the



application. We would like to see some funding allocated for this type of work.

The prime mover for funding the R & D I have described is that wind systems costing \$1000 per kW are economically attractive today as fuel savers in small diesel powered utility systems in the U.S. and abroad.

Comments by Robert E. Barrows  
WTG Energy Systems, Inc.

In regard to another area that was mentioned earlier, we need more data on the effects of airfoils in the stalled condition. As our machine is a fixed pitch machine, we would like to see what the overall effect is on the power output of the rotor with the airfoil in the stalled condition. We are getting a lot of data on that ourselves from our field testing, but we don't have any good analytical data to work with.

Also, we haven't really seen much published in regards to the effect of efficiency versus tip speed ratios. We are using about six, and some other people are using, I believe, up to 12. There should be some kind of analytical published approach specifically related to large wind turbines. During field testing of our machine, we noticed that there seems to be a stabilizing effect on the tower due to the gyroscopic effect of the rotor. If soft towers are considered, and if this stabilization factor is a benefit this will significantly effect the price of the tower. I would like to see some work done in that area.

Comments by Robert E. Donham  
Lockheed Aircraft Service Company

One of the research areas that I see was suggested by some comment made by NASA about performance measurements taken on their spar blade when compared to the existing blades for the MOD-0. It is felt that rotor performance, and possibly also low-speed airfoil aerodynamics, are activities which should be experimentally addressed. The spar blade configuration appears to be well suited to changes in airfoil shape and other geometric variables. Accurate measurements should be made of the flow field of those tested rotors, from which performance correlations can be established. I would like to have a better understanding and predictability of the aerodynamic performance because of its strong bearing on economic viability.

Another item that surfaces was the question of icing protection. We are not too worried about it now, but there will probably be problems in the future. It would be well to look at this question.

The last item I have deals with the importance of full-scale testing. Actually, NASA has the best windmill computer in the United States, that is, the experimental facilities in Sandusky. In this regard, my recommendation is that as soon as new concepts or configurations are developed, such as tubular towers, teetering hubs, etc., these elements should be dynamically modelled

and tested with full-scale equipment. Such full-scale testing gets directly at operational experience and the learning required about the actual problems associated with those ideas.

Comments by Richard R. Douglas  
Boeing Engineering and Construction Company

Fortunately or unfortunately, we just finished a year and a half of rather intensive and fairly well-funded studies of all of the concepts that we could think of and perhaps two or three hundred trade studies. We feel we have vigorously assessed each one of them. At the moment, we see no large concept changes that might have a pronounced effect on the cost of electricity on MOD-2. That is not to say that some concepts are not worth studying. Some concept changes are to be expected, but we do not see large reductions in cost of electricity.

What I do see, and it's been my experience with the Boeing Company for some time, is an evolutionary improvement. In the airplane business, when a vehicle is introduced into commercial practice, we don't build a prototype. We sell a lot of them, and then we incorporate tens of thousands of changes. For example, the 727 you buy today is not even close to the 727 that the first customer bought. It has been improved in every way. It has cost reduction and efficiencies in improvement by the tens of thousands. It is my experience that the real big gain in cost of electricity reduction, or in component cost reduction or efficiency gains, will come by a component-by-component study down into the details.

These improvements basically fit into three phases. The first is a component-by-component study that can be made as early as the completion of the design. The second phase involves changes that can be made as we understand the tooling requirements at the fabrication phase. There are a rather large number of changes in this category. Then, by far the largest number of changes are made from the feedback, actual maintenance and field experience, from a large number of units in the field, all of which leads to product improvements. Thus, I feel the best approach is a component-by-component study, although that is not to say we won't follow up concept changes that are produced.

Comments by Lou Mirandy  
General Electric Co.

We have three items to discuss for R & D research. The first is in the area of experimental investigation, which should be along the lines that NASA has instituted with the MOD-0 program. This machine is evaluating such things as low cost blades, teetering, and a soft tower. With the new MOD-1 going on line, similar programs could be established on a larger size wind turbine to learn more about what new improvements would do in actual operation. For example, a fiberglass blade will be tested on this machine. The investigation of innovative ideas on wind turbines such as the MOD-1 should be encouraged. It is especially important that the test beds that we have now be used to check out advanced concepts.

The second area that we think should be looked at further is the kind of work done by Southern California Edison, who have taken the bull by the horns in a commendable fashion, and placed a wind turbine in operation by themselves. Since the utilities will be the eventual users of these machines, we feel that the government should encourage this effort and support other utilities to enact similar programs.

Finally, since the trend in many of these large machines seems to be going towards composite blades, we think that perhaps more basic research work should be done in composites.

Comments by Carl Zweben  
General Electric Company

Composites have a lot to offer for blades because of their stiffness characteristics. The blade is primarily loaded in one direction with some torsional stiffness requirement, and that makes composites well suited for this application. Composites also have good fatigue resistance and damage tolerances, which are also important.

The potential for low-cost manufacturing has been demonstrated, although new techniques are obviously going to be required for the wind turbine blade, which is a very specialized structure, especially in the larger sizes. The growing use of composites in helicopter blades is evidence that the materials have merit for this type of application.

My observation is that the approaches that have been used or can be tried now are fairly limited. It is therefore a good time to take a broader look at the materials and processes that are available, or could be adapted with a minimum amount of effort to large blades. Now is the time to develop the kind of technology that will be required for the next generation of blades, so that costs can be reduced.

Comments by Glidden S. Doman  
Hamilton Standard

There aren't many items that I can add to what's been said so far. I would just like to state my general agreement with the Boeing point of view as was expressed by Dick Douglas. The essential immediate task is to get the development engineering process going thoroughly and successfully. As for research recommendations, although it may sound strange, I think the field lacks cost knowledge. A data/experience bank on cost and weights is needed. It is very difficult to conduct a trade study or a system improvement effort with the guesswork involved in the cost of some system elements. We are doing well, but I don't think we are doing as well as we could to get all the information organized. Perhaps there is something that NASA-Lewis can do to help to get the cost projection "art" converted to somewhat of a science.

The other suggestion relates to analysis codes. I think there is still some work that should be done on code verification. Indications are that we

are highly content with our computer codes and we are satisfied with what they do for us. The fact is, however, that we stay two or three jumps ahead of the engineering needs. We are so busy using them successfully, that they are not challenged very often. We should sponsor a very energetic code verification effort in which we can all participate. Otherwise, project people will move along and neglect it. I don't think we are headed for any trouble in the machines that are about to be assembled, but we could have avoided some embarrassments and some project delays if we had really concentrated on code verification. Perhaps it can be done in a wind tunnel with more instrumentation and model work. It probably would cost a million dollars, but my personal opinion is that it would pay.

Comments by David A. Spera  
NASA Lewis Research Center

There are two specific R & D items I wish to address. First on composites. Composites require considerable money for development at the beginning, and money must be planted now if a composite blade is to be reaped two or three years from now. Thus, I readily affirm a need for R & D funding, and probably very heavy R & D funding, for composite blades.

Secondly, the prediction of wind speed and its duration during the year at elevations of 150 to 400 foot or more is a very difficult job. I don't believe that we can currently substantiate the total number of kilowatt hours predicted for the output of a machine over the period of a year. We have the technology for doing that, but it has to be calibrated somehow.

Comments by Robert Thresher  
Oregon State University

As was mentioned throughout the conference, the goal of the federal program is now 2 to 3¢ per kWh, up from the earlier 1 to 2¢ per kWh. This should greatly relieve the R & D requirements, and perhaps extensive R & D is not as critical now as it would have been with a 1 to 2¢ goal. However, even with that change, there is one area that hasn't been mentioned a great deal. That is the statistical input from the wind in terms of direction and magnitude. Even the shear that we design for is time dependent in nature, and it may be completely reversed at various times from what we suppose. The thought is that it may be possible to decrease system weight if some analytical technique can be developed that might account for the statistical nature of the wind and the resulting fatigue loads.

Comments by Sven Hugosson  
National Swedish Board for Energy Source Development (NE)

As far as design drivers are concerned, I believe they should be to:  
(1) reduce component weight; and (2) aim for dynamic uncoupling of the system. For weight reduction, the principal areas are the rotor blades and the nacelle. For uncoupling, the paths of interest are: rotor → drive train → tower for the shaft torsion mode; rotor → tower for the tower torsion mode; and rotor → tower → ground for the tower bending modes. These design drives then lead to the requirements of lightweight blade materials, integrated machinery, and detailed structural dynamic analyses.

These required developments call for R & D in the following areas: First, for research, there is: wind data (profile and turbulence); materials (fatigue properties); and WECS-network interaction (control and transmission). Secondly, in the category of applied development, we should use analytic methods, evaluate existing units, and develop improved designs. Principal items are: dynamic concepts; control concepts; hub types; blade construction; power generation (low rpm generators, frequency converters); and maintainability.

Comments by John T. Parsons  
Robinson Industries, Inc.

An item of cost that has been touched upon only barely by two speakers is that of product liability insurance. One would normally expect that such a cost might be small, and that it might be based on rating and experience. It is of interest to note that after building 40,000 helicopter blades and after overhauling 140,000 blades, there were only two claims. One claim was for \$10,000, and one claim required the insurance company to spend about \$50,000 in defense. However, after this, the product liability charge was increased to 11 percent of the sales dollar. Thus, when you talk about cutting \$40,000 or \$60,000 off the cost of a blade, you had better get your arguments together before you call in an insurance company for a rating. The story has to be complete, and you had better have evidence.

It is also significant to note that somewhere around 1957 or '58, we were sued for \$456,000 for the death of three men in a Navy-owned helicopter for which we built the blades. It was an injury trial suit. The jury awarded zero damages because we were able to prove, by our quality control records, the good condition of the blades when they left our plant, and we were able to follow the history of those blades during the intervening years.

Incidentally, for those of you who are thinking of using low carbon steel, the blades that failed had a 4130 steel spar. The origin of the failure was a pit 0.0015 inch deep.



## Comments on Low-Cost Megawatt Size Wind Turbines

Ronald L. Thomas and Richard M. Donovan  
NASA Lewis Research Center

This presentation deals with considerations of advanced technology for reducing the cost of electricity of megawatt size wind turbines to less than 3¢/kWh. The elements that make up the cost of electricity (COE) value for large wind turbines are given in figure 1. Also shown is the goal COE value for advanced systems, and the COE value for one hundred Mod-2's. The approach to meeting the cost goal for the advanced MW Wind Turbine are outlined in figure 2. The Mod-2 wind turbine 100th unit cost estimates were used as the representative state-of-the-art technology to establish the baseline for large machines.

The sensitivity of COE to changes in each of the COE parameters is shown in figure 3. The black dot in the center represents the Mod-2 100-unit baseline design (values in fig. 1). The dashed line represents the goal of 2.5¢/kWh. The solid lines below the baseline point show the percent increase or decrease in each parameter necessary to go from 3.5¢/kWh to 2.5¢/kWh. For example, the machine capital cost would have to be reduced 30 percent or the annual energy increased 40 percent to meet the target COE. It is interesting to note the relative insensitivity of O&M on COE. It is fortunate that O&M costs are small compared to the capital costs, since the O&M estimates are the most difficult to predict and to verify. Figure 3 also shows the powerful effect of mean wind speed at the site ( $\text{kW} \propto V^3$ ).

Inasmuch as large percent reductions in capital costs might be more attainable than corresponding percent increases in energy capture, component costs should receive close scrutiny. The Mod-2 100th unit costs and weights are summarized in figure 4 for each of the major subassemblies. The total cost is estimated at \$1.7M, the weight at 588,000 pounds and the total average dollars per pound is \$2.92.

Figure 5 summarizes the COE goals for each major subassembly of the advanced MW WT. In the figure, the Mod-2 COE for each subassembly is compared with the advanced MW WT and the percent reduction required in each subassembly to achieve the overall COE goal is also shown.

The objective for the rotor subassembly in particular is to reduce costs by 50 percent or to increase energy capture by 10 percent. Calculations indicate that a 10 percent increase in rotor energy capture has the same impact on WT COE as reducing the cost of the rotor by 50 percent! This shows the importance of investigating all aerodynamic techniques that will increase energy a percent or so. Several small aerodynamic improvements combined together can have the same impact as a major fabrication improvement.

This presentation will discuss in detail only the rotor subassembly with regards to possible cost reductions and/or increased energy capture. Figure 6 summarizes the objective and outlines the approach for the rotor subassembly. The investigations of blade designs, materials and fabrication techniques that have potential for lower costs have been discussed at great length at this workshop. It appears several designs are available that should result in lower cost rotors than now available. These will not be discussed here since they have already been presented.

The second area for possible rotor cost reduction is that of different concepts. These include ideas such as a one-bladed rotor and the use of a flap-spoiler in place of a tip control. The advantages and disadvantages of a one bladed rotor are summarized in figure 7. The Mod-0 test bed is being re-configured for testing and evaluation of the one-blade concept in the Spring of 1980. The use of a flap-spoiler is also being researched, and this concept will also be tested and evaluated on the Mod-0 in 1980.

The third item, increased energy capture, appears to offer considerable potential. Figure 8 lists several ideas for increasing energy capture and the estimated percent increase. Most of the ideas have potential for a few percent increase in energy, except for the multi-speed rotor operation which has potential for 10 to 20 percent.

Figure 9 compares the performance of the single speed rotor baseline Mod-2 with a conceptual variable speed Mod-2 and a two-speed Mod-2. Calculations were made for two cases: maintaining rated power; and maintaining rated torque. The figure shows that there is much to be gained by letting the generator increase in rating while keeping the torque at the baseline rated value by increasing the rotor RPM. This simple investigation showed that a 2-speed Mod-2 could achieve a 10 percent increase in energy capture. The completely variable concept shows a potential for 20 percent, but this is not



realistic because of drive train efficiencies and potential rotor/machine dynamic problems.

The WT power output versus wind speed is shown for the Mod-2 single-speed baseline and the variable speed concepts in figure 10. It can be seen that the variable speed increases the power output at the lower and higher wind speeds and allows for much higher rated power for the same design torque. The power output versus wind speed for the two-speed Mod-2 concept is shown in figure 11. For this concept, the speed ranges were arbitrarily picked at a 2 to 1 ratio of 24 and 12 RPM. Figure 11 shows that the two-speed concept provides higher power at the low and high wind speeds and allows the generator rating to be increased from 2500 KW to 3400 KW. However, the power output for the two-speed concept is less than the single speed design at wind speeds of 15 to 20 miles per hour.

Figures 12 and 13 show the Mod-2 rotor performance in terms of coefficient of performance ( $C_p$ ) versus wind speed. Figure 12 is for the one-speed baseline Mod-2 and figure 13 is for the two-speed concept. It can be seen from these figures how the two-speed concept keeps the  $C_p$  value higher over the range of wind speeds and thereby increases the energy capture. Also, if figure 13 were superimposed on figure 12, the potential benefits of a three-speed design become apparent.

The two-speed concept was presented here to show the potential for increasing WT energy capture. There are ways of changing the poles in the generator to keep the generator at 60 Hz even though the generator speed is changed. This allows the gearbox and drive train to operate at several different speeds without adding complexity to the drive train.

In summary, it appears that the advanced MW cost goals, though challenging, can be met with advanced technology. This technology such as multi-speed operation, should result in lower weight, and lower cost machines that operate more efficiently and capture more energy.

## WIND TURBINE COST OF ELECTRICITY

$$\text{COE} = \frac{(\text{CAPITAL COSTS}) \times (\text{FIXED CHARGE RATE}) + 2 (\text{ANNUAL O\&M})}{\text{ANNUAL ENERGY}}$$

COE GOAL:

2.5¢/kW/H

FOR MOD-2:

$$\text{COE} = \frac{(\$1.717 \times 10^6) (0.18) + 2 (\$15,000)}{9.75 \times 10^6 \text{ kWh}} = 3.45\text{¢/kWh}$$

FIGURE 1

## APPROACH TO MEETING ADVANCED MW WIND TURBINE COST GOALS

- USE THE MOD-2 100<sup>TH</sup> UNIT DESIGN AND COSTS AS REPRESENTATIVE OF STATE-OF-THE-ART TECHNOLOGY FOR LARGE WIND TURBINES.
- ESTABLISH COST GOALS FOR EACH MAJOR ASSEMBLY (DRIVE TRAIN, ROTOR, ETC.) AND COST ITEM (ASSEMBLY, INSTALLATION, ETC.) THAT WILL RESULT IN REDUCING CAPITAL COSTS.
- ASSESS ALL POTENTIAL MEANS FOR INCREASING ANNUAL ENERGY CAPTURE.

FIGURE 2

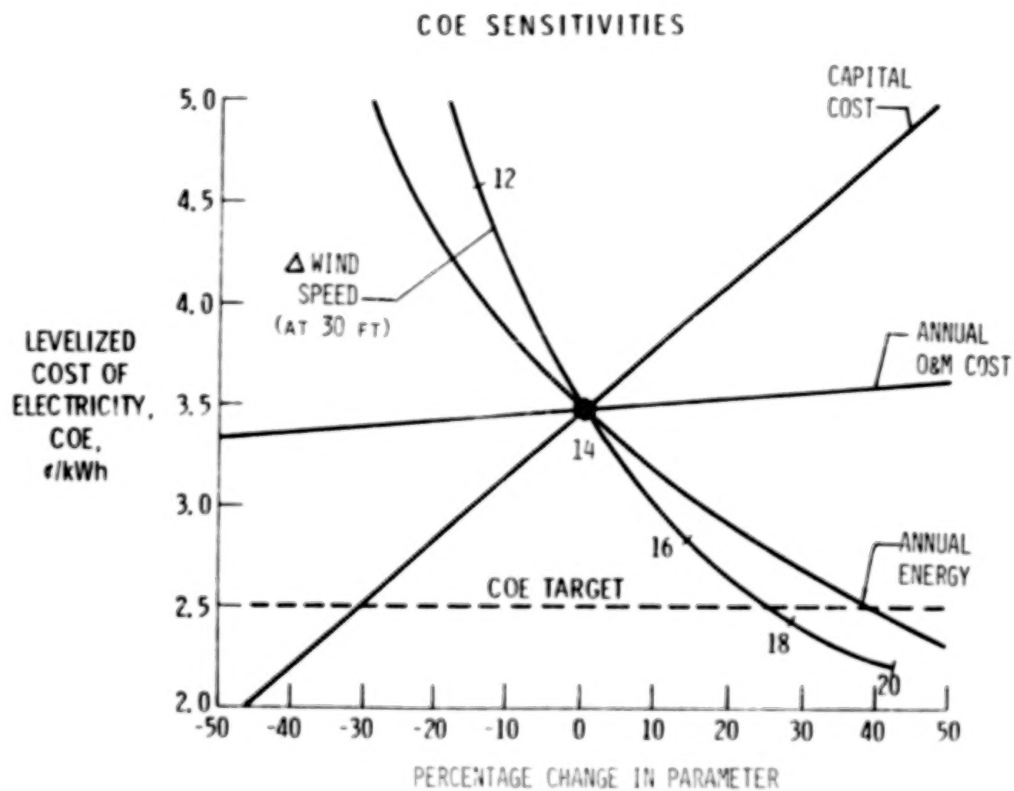


FIGURE 3

**MOD-2 COSTS AND WEIGHTS**

SUBASSEMBLY	\$K	LBS., K	\$/LB
DRIVE TRAIN	379	104	3.65
ROTOR	329	170	1.94
TOWER	271	251	1.08
NACELLE/YAW	184	63	2.92
SITE PRE/FOUNDATION	162	---	----
INSTALL/ASSEMBLY/CHECKOUT	137	---	----
SPARES	70	---	----
TRANSPORTATION	29	---	----
	<hr/>		
	\$1561	588	2.65
10% FEE	156		.27
	<hr/>		
	\$1717		2.92

FIGURE 4

# ADVANCED MW WT COE GOALS

	MOD-2 ¢/KWH	% TOTAL COE	GOAL, % REDUCTION	ADV. MW ¢/KWH
DRIVE TRAIN	0.69	20	30	0.48
ROTOR	0.60	17	*	0.30
TOWER	0.50	14	20	0.40
NACELLE/YAW	0.33	9.5	30	0.23
SITE PREP/FOUNDATION	0.30	8.5	20	0.24
INSTALL/CHECKOUT	0.25	7	20	0.20
SPARES	0.13	4	20	0.10
TRANSPORTATION	0.06	2	20	0.05
	2.86	82	--	2.00
10% FEE	.29	8	--	.20
2 x O&M	.30	10	--	.30
TOTAL	3.45	100	27.5	2.50

\* GOAL FOR THE ROTOR IS AN INCREASE OF ENERGY CAPTURE OF 10%. THIS IS EQUIVALENT TO A 50% REDUCTION IN THE COE DUE TO THE ROTOR.

FIGURE 5

## OBJECTIVE AND APPROACH FOR ADVANCED ROTOR DESIGN

OBJECTIVE - REDUCE ROTOR COSTS BY 50% OR INCREASE ENERGY CAPTURE BY 10%

### APPROACH

- INVESTIGATE BLADE DESIGNS, MATERIALS AND FABRICATION TECHNIQUES THAT HAVE POTENTIAL FOR LOWER COSTS (I.E. - LOW COST BLADE CONTRACTS)
- INVESTIGATE ROTOR SYSTEMS THAT HAVE POTENTIAL FOR LOWER COSTS (ONE-BLADED ROTOR, FLAP-SPOILER IN PLACE OF TIP CONTROL)
- INCREASE ROTOR ENERGY CAPTURE

FIGURE 6

## ALTERNATE ROTOR CONFIGURATIONS

ONE-BLADED ROTOR WILL BE INVESTIGATED

### ADVANTAGES

- ONE BLADE LESS EXPENSIVE THAN TWO (NEED COUNTER WEIGHT)
- ROTOR OPERATES AT 35% HIGHER RPM (SMALLER GEAR BOX)
- HURRICANE THRUST LOADS ON TOWER ARE REDUCED

### DISADVANTAGES

- NOT AS AERODYNAMICALLY EFFICIENT (-10%)
- MORE DIFFICULT TO STARTUP

FIGURE 7

# IDEAS FOR INCREASING ROTOR ENERGY CAPTURE

## ITEM

## % INCREASE IN ANNUAL ENERGY

- MULTI-SPEED ROTOR OPERATION	10%
- OPERATE AT HIGHER CUT-OUT VELOCITIES	1*
- IMPROVE BLADE TIPS TO REDUCE LOSSES AND INCREASE MAXIMUM $C_p$	2
- OPERATE WITH SMOOTHER AIRFOILS	3
- KEEP YAW ANGLE AT ZERO WITH PASSIVE YAW	2
- SCHEDULE BLADE PITCH FOR OPTIMUM BETWEEN CUT-IN AND RATED	2
- USE TRAILING EDGE FLAP TO INCREASE $C_p$ AT LOW-SPEED	2
- HIGHER SOLIDITY CENTERBODIES TO INCREASE $C_p$ MAXIMUM	?
- USE HIGHER L/D AIRFOILS TO INCREASE MAXIMUM $C_p$	?
	<hr/>
	22%

- \* ONLY 1% AT SPECIFIED 14 MPH SITE, BUT COULD BE 10% AT HIGH WIND SPEED SITES LIKE SAN GORGONIO

FIGURE 8

# MULTISPEED ROTOR PERFORMANCE

	RATED POWER, 2500 KW			RATED TORQUE, $10^6$ FT-LBS		
	RPM	TORQUE	OUTPUT	RPM	POWER	OUTPUT
BASELINE MOD-2	17.5	1.00	1.00	17.5	2500	1.0
VARIABLE RPM	10 - 27	.76	1.05	10 28	3800	1.2
TWO SPEED RPM	9.4, 18.8	.93	1.02	12, 24	3400	1.1

FIGURE 9

# POWER PROFILE COMPARISONS BASELINE VS VARIABLE RPM

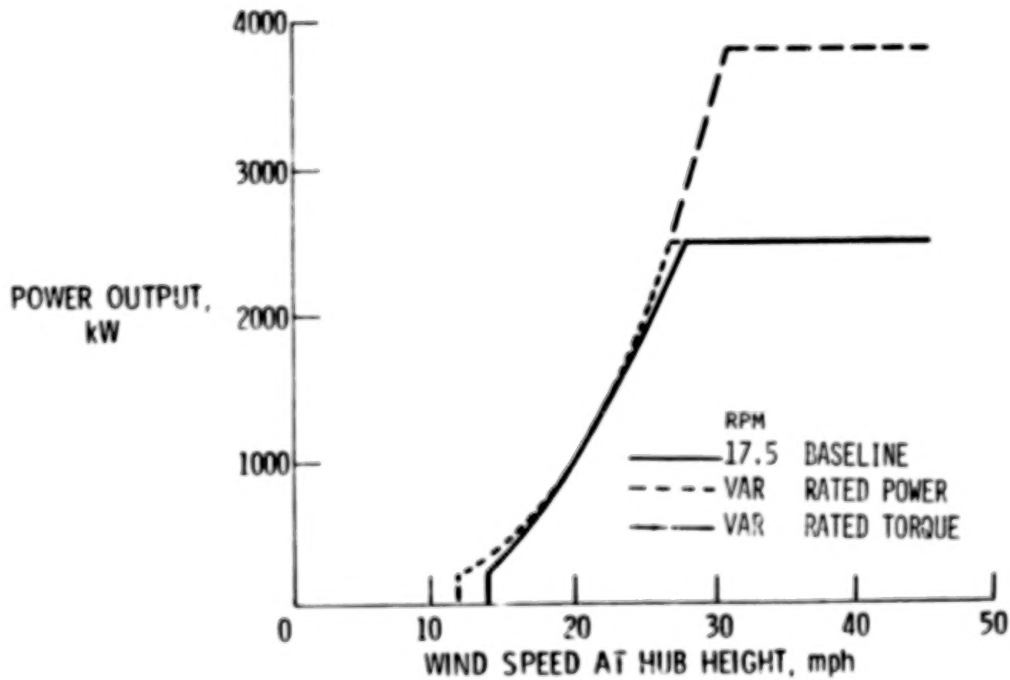


FIGURE 10

# TWO-SPEED RPM POWER PROFILE

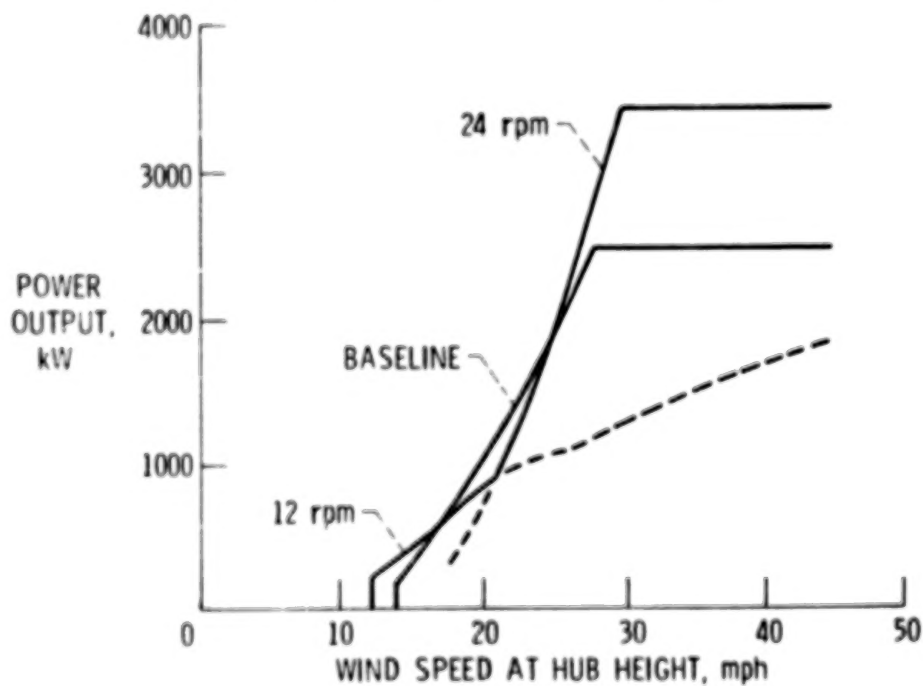


FIGURE 11



# BASELINE SINGLE SPEED ROTOR (17.5 RPM)

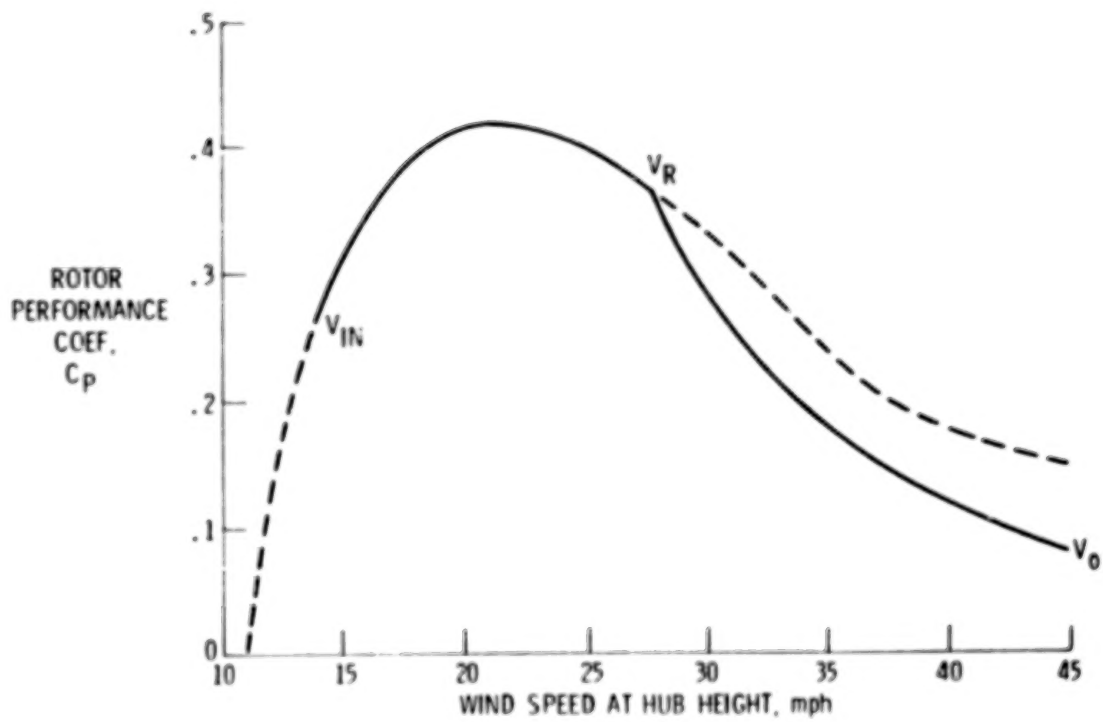


FIGURE 12

# TWO-SPEED ROTOR

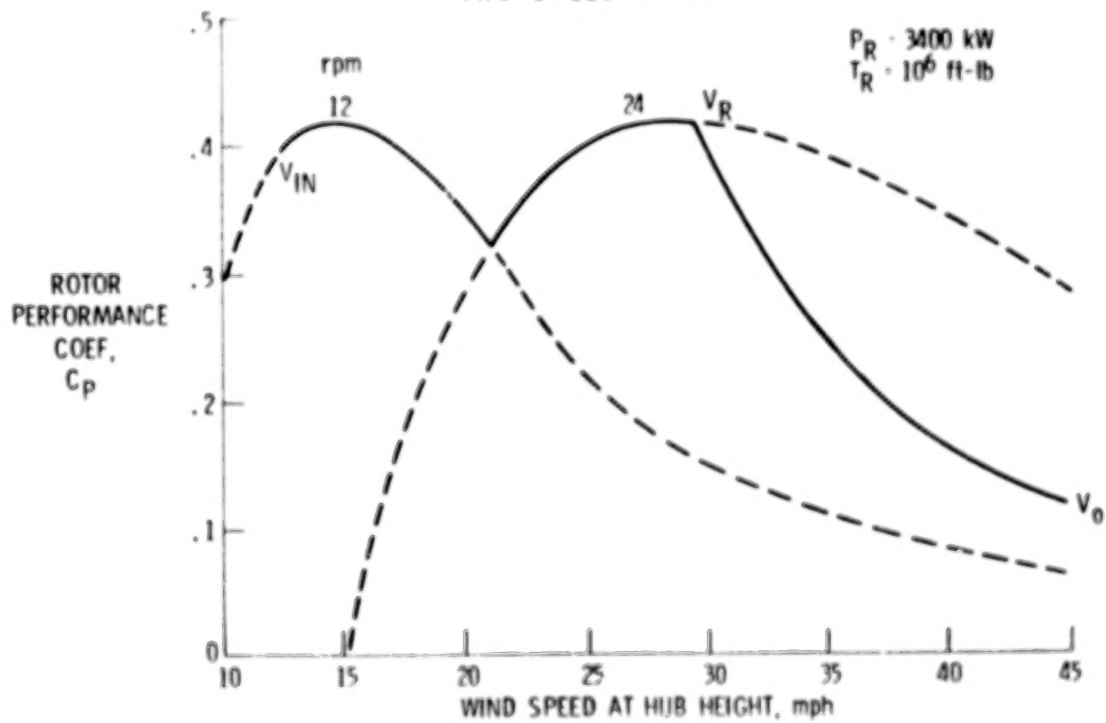


FIGURE 13

## Comments on Utilities and Wind System Economics

Edgar DeMeo  
Electric Power Research Institute

At this point, a few comments are in order relative to the economics of wind turbines from the utility point of view. It is clear that the entire issue of economics is an extremely complex one. Nevertheless, the goals that have been set for this program - 2 to 4¢ per kWh, calculated the way it was explained here - are a reasonable and sensible set of goals. I would like to try and explain why.

First of all, the value of a wind turbine in a utility system will have two parts. One part will be due to the fuel which is displaced by the wind turbine, the so-called energy value. The other part will be a capacity value, that is, a credit for some other kind of generation that can be deferred because some wind generation was installed. Of these two, in most cases and probably in all cases, the larger will be the fuel credit, that is, the energy credit. The capacity credit will be smaller. In addition, it will take several years of real experience with utility-operated wind machines before the capacity credit will actually be allowed and considered something real.

In order for a utility to establish an accurate assessment of the value of wind generation in its system, it has to go through a rather detailed simulation of the operation of the entire system, looking at all of the generating units on the system, which is fairly complex. One of the things that is very clear from this pursuit is that whatever the results are, they are very much utility specific. They vary considerably from one utility to another, depending on what kind of generation system the utility has, what kinds of fuels it burns, and what special constraints and conditions apply. There is much variation in these parameters. Several studies of this type have been done for wind, some with EPRI support, and others funded by DOE. In fact, at the workshop held in Monterey about a month ago, this entire question of economics was one of the major issues that was discussed.

Another thing that is clear about wind turbines and their interactions with the utility system is that the wind turbine will displace some or all of the fuel types that are present on the system. It won't just displace the most expensive fuel, it will displace some of everything that the utility happens to use. So, it is clear then that wind turbines will do best in those utilities which have a predominance of expensive fuels like oil or gas.

I believe that the goals which were presented here for the wind program will, if achieved, allow wind turbines to compete on the basis of fuel displacement alone, which is very important. This applies, of course, to utilities with good wind resources and high fuel costs. For example, take a utility that burns a good deal of expensive fuel like oil or gas. If that utility is interested in wind and has good wind resources, and if that utility can convince itself that the wind turbine will produce energy when units burning these expensive fuels would normally be operating, then a number like 3 to 4¢ per kWh for the wind machine is very attractive. That is the situation which, I believe, prevails in more than a handful of utilities around this country.

Furthermore, a number like 2 to 3c per kWh would be attractive to a much larger number of utilities, because this would also compare favorably with average fuel costs, that is, not just the fuel cost of the most expensive fuel used, but an average of most or all of the fuels that are used. That is why I feel the goals set for this program are reasonable.

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Question. - One of the problems that we have been confronted with ever since the start, and which ought to be addressed soon, is the control of frequency at high levels of penetration. Most of the wind turbines that NASA-Lewis is developing rely on the utility grid to control their frequency. In other words, the pitch control mechanism is really just an overspeed power limiting device. Now that utilities are considering large blocks of wind turbines at potentially high penetration levels in synchronous lockup, do you feel that pitch control is an acceptable control system configuration?

Answer. - I am not sure I can comment on your question here, but I think these are the kinds of questions for which we won't really get full answers until there is more operation experience with actual hardware.

Question. - If a hundred megawatts of installed capacity comes on line in a very short time, is the utility system equipped to control that frequency?

Answer. - That will depend on how quickly the utility can change the level of output from other generating units on the system. Again, that is something that will vary considerably from one utility to another. This question could be asked of one of the other speakers who might have given more thought to this matter.

Question. - You stated that the cost of energy now of 2 to 3c per kWh is the average for the utility industry across the country. The utility industry is made up of a lot of old plants. Would you care to comment on the cost of energy now for a newly installed coal-fired plant, and a newly installed oil plant?

Answer. - It varies all over the lot. But generally a number that corresponds to 3c per kWh for a new coal-fired plant would be pretty good. The fuel components of that cost would be maybe half of that total.

Question. - Are there any reports or documents available to provide a feel for the cost of energy when there is no capacity credit, that is, for what's going on in the country right now?

Answer. - Yes. EPRI puts out a periodically updated technology assessment guide which lists much of the type of information desired. There are also several other documents which come out every year that look at the energy supply and demand situation. As for the economics of wind integration into utility systems, we do have a report available on the first project of this type that we have funded. The work was done by General Electric, Schenectady, New York. I have a copy of the summary document from this report, and if anybody would like to obtain a copy, let me know.

## CLOSURE

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I would like to go back a few years to the last workshop, which was held in Washington and reflect on my observations of the differences between that workshop and this workshop. The basic difference is that 2 years ago almost everybody was talking primarily about plans, and some people were in the early design phases of their wind turbines. As has been shown here in these last 3 days, we have gone well beyond the planning stage. Wind turbine development has made major progress in both government and private programs.

We now have large-scale hardware either deployed or about to be deployed. My experience in R&D, and with due apologies to the analysts, is that the hardest job in the world is to get a reliable piece of hardware to perform the way it should. We at NASA believe that we and the organizations that work with us will be facing our greatest challenge in the next 2 years. We have already faced a primary challenge at Clayton. We know now that the basic design approaches we are following are sound. But, there is a long way yet to go in terms of demonstration hardware. The primary thrust for all of us is to get the hardware working as we think it should.

1. Report No. <b>NASA CP-2106</b>		2. Government Accession No.		3. Recipient's Catalog No.	
4. Title and Subtitle <b>LARGE WIND TURBINE DESIGN CHARACTERISTICS AND R&amp;D REQUIREMENTS</b>				5. Report Date <b>December 1979</b>	
				6. Performing Organization Code	
7. Author(s)				8. Performing Organization Report No. <b>E-290</b>	
9. Performing Organization Name and Address <b>National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135</b>				10. Work Unit No.	
				11. Contract or Grant No.	
				13. Type of Report and Period Covered <b>Conference Publication</b>	
12. Sponsoring Agency Name and Address <b>U.S. Department of Energy Distributed Solar Technology Division Washington, D.C. 20545</b>				14. Sponsoring Agency Code Report No. <b>CONF-7904111</b>	
15. Supplementary Notes  <b>Editor, Seymour Lieblein, Technical Report Services, Rocky River, Ohio.</b>					
16. Abstract  <b>A workshop on Large Wind Turbine Design Characteristics and R&amp;D Requirements was held at the NASA Lewis Research Center in Cleveland, Ohio, April 24-29, 1979. Both horizontal axis and vertical axis machines were covered. The sessions included the development status of large wind turbines, wind turbine blade design characteristics and operating experience, and special topics. Working group sessions and panel discussions in the major topic areas were also held. The proceedings of this workshop are presented herein. Included are the formal prepared papers and the edited versions of the transcriptions of panel discussions, working group summaries, and questions and answers.</b>					
17. Key Words (Suggested by Author(s))  <b>Wind power; Wind turbine; Horizontal axis and vertical axis turbines; Design characteristics; Technology status</b>			18. Distribution Statement  <b>Unclassified - unlimited STAR Category 44 DOE Category UC-60</b>		
19. Security Classif. (of this report)  <b>Unclassified</b>		20. Security Classif. (of this page)  <b>Unclassified</b>		21. No. of Pages  <b>464</b>	
				22. Price*  <b>A20</b>	



**END**

APRIL 27, 1981